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EVALUATION OF HEAT ENGINES FOR HYBRID VEHICLE APPLICATION

By
H. W. Schneider

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Jet Propulsion Laboratory
Pasadena, California

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Evaluation of Heat Engines for Hybrid Vehicle Application

H.W. Schneider

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ABSTRACT

The status of ongoing heat-engine developments, including spark-ignition, compression-ignition, internal-combustion, and external-combustion engines is presented in this report. The potential of engine concepts under consideration for hybrid vehicle use is evaluated, using self-imposed criteria for selection. The deficiencies of the engines currently being evaluated in hybrid vehicles are discussed. The study focuses on recent research with two-stroke, rotary, and free-piston engines and concludes that these engine concepts have the most promising potential for future application in hybrid vehicles. Recommendations are made for analysis and experimentation to evaluate stop-start and transient emission behavior of recommended engine concepts.

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EXECUTIVE SUMMARY

A. INTRODUCTION AND SCOPE

The objective of the U.S. Department of Energy (DOE) Electric Hybrid Vehicle (EHV) Program is to evaluate the potential of electric vehicles (EV) and/or hybrid vehicles (HV) to significantly reduce petroleum consumption. The Jet Propulsion Laboratory (JPL) supports the DOE by performing EHV Systems research and development (R&D). In this role, JPL is responsible for the development and evaluation of hybrid-vehicle concepts. Previous HV activities included JPL assessments (References 1 and 2) and a DOE/JPL contract to General Electric Company (GE) to demonstrate HV feasibility through development of a complete vehicle (References 3 and 4). A limitation of previous assessments and vehicle development activities was that only off-the-shelf engines were considered.

Having demonstrated the technical feasibility of the electric/gasoline engine hybrid-vehicle concept (see Reference 3), JPL initiated a small-scale study to evaluate advanced engines by assessing their attributes when applied to unique HV requirements. Based on a literature search and personal communications with a variety of sources that are actively engaged in heat-engine R&D work, this study evaluates the potential of advanced heat-engine concepts for future hybrid application, using a system-prioritized rating criteria (Table 1) that builds on the findings established in the aforementioned references. In addition to fuel efficiency and emissions, these criteria emphasize stop-start capability, low motoring-power requirements, weight, packaging flexibility, service needs, and accessibility. While most of the criteria are conventional, added emphasis has been placed on weight and packaging flexibility as a result of the experience gained during the GE contract to build an HV. In addition, a recent assessment of HVs (Reference 5) indicates the strong influence of these criteria to the viability of the HV.

Included in the evaluation (Table 2) are advanced versions of internal-combustion, four-stroke, spark-ignition piston engines; Wankel-type rotary engines; spark-ignition, two-stroke piston engines; and external-combustion engines such as Brayton (gas turbine) and Stirling engines. Production spark-ignition piston and indirect-injection diesel engines of current design are included for comparison. The status and potential of direct-injection, high-speed diesel engines and free-piston engines are briefly discussed. All of the engines evaluated are atmospherically aspirated. Turbocharged engines were not considered because of their incompatibility with rapid stop-start operations (lag) and because they are too costly for hybrid applications in which the heat engine is only an accessory.

The original task plan provided for an in-depth, fact-finding follow-up to fill information gaps and for an analytical drive-cycle evaluation of the most promising candidate heat-engine concepts. This portion of the task could not be completed because funding was discontinued beyond FY 1983.

Table 1. Priority Rating of Hybrid-Vehicle Engine Properties

Powertrain Concepts	Separate System		In-Series	Parallel
	Primary Electric Power	Primary Heat-Engine Power	Primary Electric Power	Combined Electric Heat-Engine Power
Heat-Engine Properties	Auxiliary Heat-Engine Power	Auxiliary Electric Power	Heat-Engine Battery Charging	
Low Brake-Specific Fuel Consumption	2	5	4	4
Low Brake-Specific Emissions	3	5	3	4
Low Weight	3	4	3	3
Packaging Advantages	4	4	4	5
Low Cost	4	4	4	4
Low Service Needs	5	3	5	5
Stop-Start Capability	1	1	3	5
Instant Full-Power Capability	3	3	3	5
Starting Power Requirements	Manual starting desirable	Conventional electric starter	Cranking to firing with generator	Cranking up to 50% nominal speed with motor generator desirable

Ratings: 1 - Unimportant; 2 - Less Important; 3 - Desirable; 4 - Important; 5 - Very Important

As originally planned, this heat-engine evaluation task consisted of three activities:

(1) Literature search and review of on-going R&D.

(2) Hardware evaluation.

(3) Hybrid-vehicle simulations using data from Steps (1) and (2).

This report presents the findings of the first of three subtasks.

A summary of the R&D status for various heat-engine concepts is presented at the end of this section to provide insight into the study's conclusions and recommendations. An in-depth discussion on the topics summarized below are presented in the technical discussions beginning in Section I of this report.

B. CONCLUSIONS

Figure 1 compares the relative ratings for various engine concepts against three parameters that are key for HV applications. Figure 1a shows relative fuel efficiency rankings for engine types while Figure 1b shows overall specific volume rankings, and Figure 1c compares the specific weight characteristics. From these figures it can be concluded that the unique requirements for engines in HVs make it likely that non-conventional engines offer the opportunity for substantial improvements in both performance and marketing of HVs. Furthermore, it seems that the use of off-the-shelf engines has placed too many constraints on previous HV developments and studies. The following engine-specific conclusions support the preceding general conclusions.

(1) The advanced, fuel-injected, two-stroke engine is the most attractive choice of a heat-engine concept for use in a hybrid vehicle for the following reasons:

(a) Its fuel efficiency is excellent, exceeding that of a direct injection diesel at part load (see Figure 1a) while exhibiting emissions that are acceptable without after treatment (Figure 2) and are excellent if used in conjunction with a thermal reactor.

(b) Having no valves, the advanced two-stroke engine is simple in design and its service needs and production cost are low. It is also extremely attractive from the packaging standpoint because it occupies only one-third of the volume (see Figure 1b) and is more than 50% lighter (see Figure 1c) than a conventional or advanced four-stroke engine of comparable performance.

Table 2. Comparison of Engine Design and Performance Criteria

Engine Concept	Internal Combustion Engines						External Combustion Engines	
	Naturally Aspirated Four-Stroke Spark-Ignition Engine			Naturally Aspirated Two-Stroke Spark-Ignition Engine		Diesels		
Criteria	Piston		Rotary (Wankel)			Indirect Injection[a] Naturaly Aspirated (Turbocharged)	Brayton	Stirling
Engine Designation	Audi 1700	May	KKM-871	ATAC	Suzuki	Rabbit	GT-100	Mod I
Developer, Researcher	Audi/WV	Ricardo	NSU	NCAL	OEC	VW	DDA	MTI
Status	Production	R&D	R&D	Research Engine	Research Engine	Production	R&D	Research Engine
Maximum Power Output, kW/rev/min	55/5000	88/6000	121/6500	16/4500[a]	80/6000	37/5000	75/68000	54/4000
Displacement, (No. Cylinders, Rotors[b])	1.7(4)	2.0(4)	3.0(2)	0.372(1)	1.2(3)	1.5(4)	-	-
Compression Ratio	8.4	15	9.5	7.5	N/A	22	4.5 PR	17.5
Maximum Power BSFC, g/kWh	312	265	380	306	300	315	200	304
Opt. BSFC, g/kWh/% power	268/63	240/67	305/35	239/67	248/75	260/30	3200/<100	225/60
Power Concentration, kW/	32.0	44	40.4	42.7	66.3		-	-
Relative Weight to Power Ratio[a]	1.0	0.78	0.80	N/A	0.4	1.2(1.05)	1.0 [a]	3.54
Relative Volume to Power Ratio[a]	1.0	0.72	0.38	N/A	0.32	1.4(1.0)	<1.0	1.64
Speed Capability, kW x rev/min	11,978	16,812	25,517 (70,000)[a]	11,339	17,925	10,498	N/A	N/A
Remarks	Continuous Fuel Injection			Pulse Fuel Injection				
	Currently in use in experi-mental hybrid vehicles	Highly turbulent combustion	Stratified combustion Dual fuel injection Dual ignition Prediction[a]	Residual gas controlled Fuel injection Unthrottled operation	Stratified dual vortex combus-tion Pneumatic fuel injection	Direct injection improves BSFC and power output by 15%	2350°F turbine inlet temperature 2-shaft regenerated Low NOx combustion including gears[a]	15 MPa Hydrogen (2177 psi)

[a]Relative to conventional low-compression-ratio, four-stroke, spark-ignition engine.
[b]Rotary (Wankel) Engine.

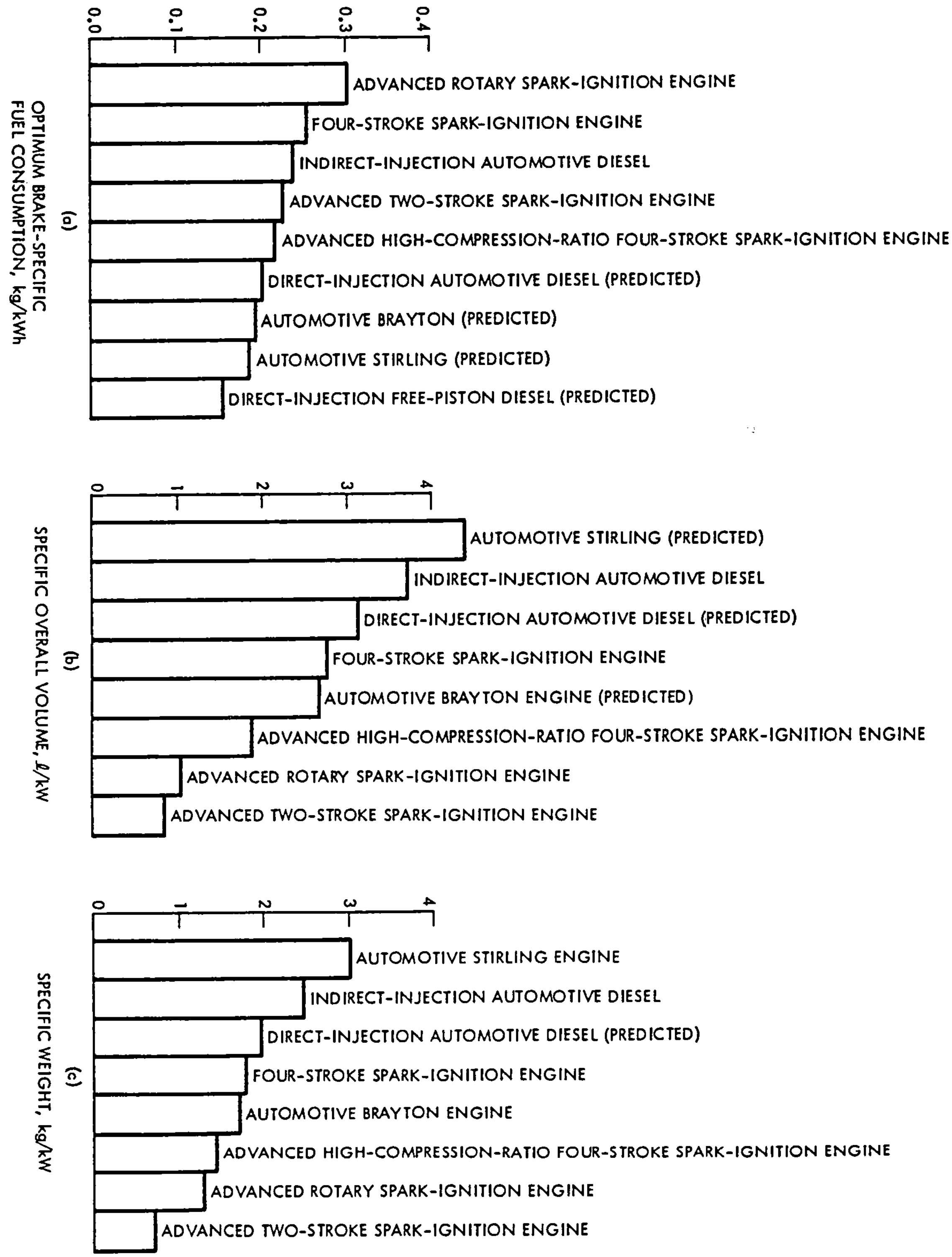


Figure 1. Comparison of Automotive Engine Design Criteria

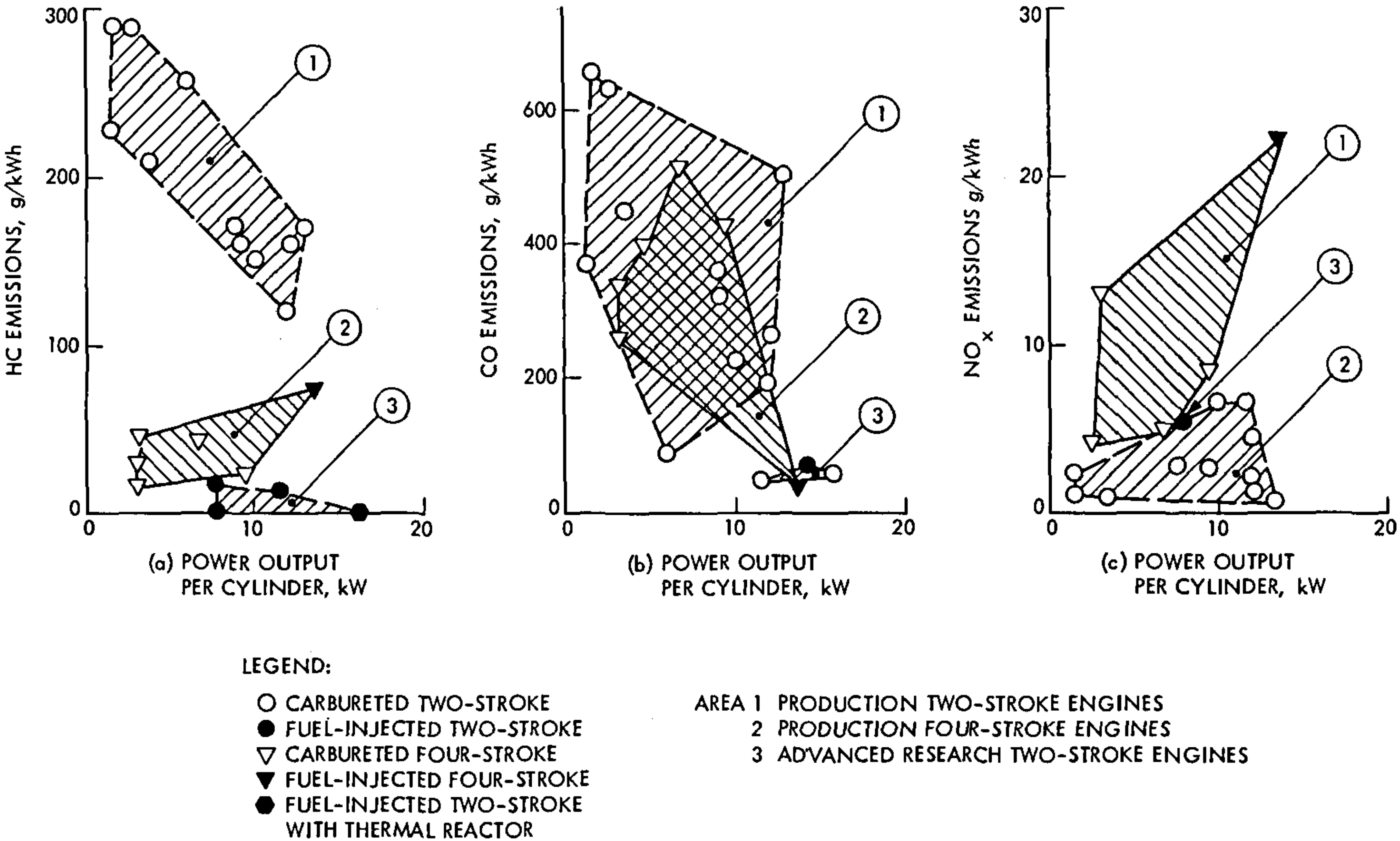


Figure 2. Emission vs. Power Output per Cylinder

(c) The stop-start capability of the advanced two-stroke engine is excellent, delivering full power instantly without a warmup. Because of its low compression ratio (7.5:1) and the absence of valve mechanisms, its motoring power is less than half that required for four-stroke engines, which translates into low transient energy losses and improved vehicle fuel economy. In conjunction with an advanced load-dependent metered injection of the lubricant, its low compression ratio (7.5:1) translates also into a long service life under stop-start operating conditions. Combining this low compression ratio with a high degree of internal turbulence allows this engine to be tolerant of poor fuel quality (octane).

(2) The (Wankel-type) rotary engine should also be considered for hybrid-vehicle application, despite the fact that its fuel efficiency is not competitive with other candidate engines (see Figure 1a) because its packaging advantages (see Figure 1b) are excellent. Its high speed capability (see Table 2) would permit it to be coupled directly to and/or integrated with a high-speed alternator. In this case, the elimination of the gear losses would compensate for its lack of fuel efficiency when compared with alternatives that must be geared to the alternator.

(3) Advanced high-compression, four-stroke piston engines have good fuel efficiency (see Figure 1a), but they are expected to have NO_x-emission problems because of higher process temperatures and impaired catalytic converter efficiency. In hybrid stop-start operation the catalytic converter does not reach its full steady-state efficiency (see Reference 3). The motoring power requirements of high-compression-ratio engines are estimated to be 50% higher than those of current four-stroke engines and 300% higher than those of advanced two-stroke and rotary engines. This fact will reduce their otherwise excellent fuel efficiency in stop-start operation. The effects of stop-start operation on engine service life will also be more pronounced with a high-compression-ratio piston engine.

(4) Free-piston, spark-, and compression-ignition engines (see Figure 1a) are attractive in terms of fuel efficiency, packaging, and vibration when coupled with a linear alternator for battery charging. Because of their capability to produce high compression ratio at low friction losses, they are well suited for the implementation of diesel combustion and of spark-ignited combustion with the use of alcohols or ammonia. If linear alternators show potential for HV application, additional development of free-piston engines should be considered because of their ability to enhance fuel economy.

(5) Based on established criteria, external-combustion Brayton (gas turbine) and Stirling engines have no potential for most hybrid-vehicle applications because of their inherent inability to be used

in rapid start-stop operation. Because of small-size effects, Brayton engines cannot be considered for power outputs below 75 kW; otherwise, they would have a potential for battery charging.

(6) The indirect-injection diesel loses out against other candidate engines in all areas (see Figure 1), including that of fuel efficiency. With direct injection, the diesel will be the most fuel-efficient, internal-combustion engine available when compared in terms of grams per kilowatt hour. However, because of its need for preheating prior to start and its high cranking power requirements, the diesel engine needs to idle instead of being turned off. Although its idle fuel-flow rate is only 20 to 25% that of spark-ignition engines, this requirement will reduce its actual fuel efficiency to some degree in most hybrid applications. The idling diesel is an attractive alternative because of hybrid accessory power and heating, which are still unresolved problems with heat-engine stop/start operation.

C. RECOMMENDATIONS

Because of the great potential for improvements in HV fuel economy, weight, and other factors affecting overall HV viability, further research is recommended. The following efforts consider the unique HV requirements in a prioritized sequence:

(1) Based on available information and the conclusions drawn by this study, it is recommended that heat-engine development be pursued as indicated in Table 3, which is grouped by hybrid powertrain concepts. The two-stroke engine is the primary choice of all powertrain concepts considered, with the exception of a primary heat-engine-powered hybrid car with an auxiliary electric powertrain that requires more than 50 kW of heat-engine power. The low-compression, four-stroke, spark-ignition engine is the best backup choice if it is equipped with a fuel-injection system and accessories developed to cope with the special requirements for stop-start operation. The free-piston diesel and the alcohol-fueled, spark-ignition, free-piston engines have an attractive potential for battery charging with a linear alternator. This potential should be re-evaluated, taking the latest technology and innovative concepts into account.

(2) The study further recommends follow-on analysis and experimentation with candidate engine concepts that are believed necessary to support the selection of a heat-engine concept for future hybrid vehicles. This effort should concentrate on the determination and evaluation of properties of specific interest to HVs that usually cannot be supplied by the heat-engine developer. The rationale and objectives of these recommendations are outlined in detail in Section V of this report.

Table 3. Engine Concepts Rated Most Promising for Hybrid-Vehicle Application

Powertrain Concepts	Separate Systems		In-Series	Parallel
Recommendations (in order of preferance)	Primary Electric Power	Primary Heat Engine Power	Primary Electric Power	Combined Electric Heat-Engine Power (Power Blending)
	Auxiliary Heat Engine Power	Auxiliary (Urban) Electric Power	Heat-Engine Battery Charging	
Heat-Engine Concept	Carbureted Two-Stroke	Fuel-Injected High-Compression Four-Stroke	Fuel-Injected Two-Stroke[a]	
			Fuel-Injected Rotary[b]	
	Carbureted Low-Compression Four-Stroke		Direct-injection diesel or alcohol-fueled spark ignit-ion, free-piston engine[d]	Fuel-Injected Low-Compression Four-Stroke[c]
Cooling System	Forced Air	Liquid		
Starting Method	Electric Starter Manual starting capability desirable	Electric Starter	Hybrid Motor/Generator, Alternator[c]	
Exhaust After Treatment	None	Two-Stroke and Rotary: Thermal Reactor[e] Four-Stroke Piston Engine: Catalytic Converter Diesel: None		

[a]For power outputs up to 50 kW.
[b]Where a direct coupling with alternator is applicable.
[c]Equipped with stop-start proof lubrication and fuel-injection system.
[d]Free-piston start: Electric excitation into boot-strapping resonance or pneumatic latch-release starter system.
[e]Waiver can possibly be obtained for hybrid with low-emission transients.

D. SUMMARY OF CURRENT HEAT-ENGINE RESEARCH AND DEVELOPMENT

1. Four-Stroke, Spark-Ignition Piston Engines

Concentrated efforts are being made by well-known developers to improve the fuel efficiency of the atmospherically aspirated, four-stroke, spark-ignition engine by raising its compression ratio and introducing highly turbulent, squish-flow agitated combustion to inhibit knocking. Fuel efficiency improvements of up to 15% and increased power output per unit displacement of up to 30% have been demonstrated. Developers are hopeful that high-compression engines can be marketed by 1985 as an option. Flame erosion of the pistons, closer head tolerances, and associated higher production cost are the pacing problems to be resolved.

2. Four-Stroke, Spark-Ignition (Wankel) Rotary Engines

State-of-the-art, rotary, four-stroke, spark-ignition engines are equivalent to four-stroke piston engines of current low-compression (8.5 to 8.7:1) design in fuel efficiency and emissions. This has been achieved primarily by improving the rotor seals, raising the compression ratio to 9:1, and introducing stratified combustion brought about by dual fuel injection and ignition. However, with these measures the fuel efficiency potential of rotary engines is essentially exhausted, with rotor sealing and combustion chamber configuration (high surface to volume ratio) being the primary limiting factors. They are, therefore, incapable of competing with high-compression four-stroke piston engines in the future. A certain potential still exists to improve the already excellent speed potential of rotaries (see Table 2) by means of lift-off seals, which still remain to be developed. The major developers of advanced fuel-injected, rotary (Wankel) engines (Audi-NSU, and Curtiss-Wright), have reassessed their goal of (some day) dominating the automotive market and are now looking for other fields of application where the packaging advantages, high-speed capability, and smoothness of the rotary concept would be of overriding importance. According to dealer information, Mazda will market a fuel-injected rotary engine as an option in 1984 model cars.

3. Diesel Engines

The uncertainties associated with diesel emission legislation in Europe and the United States as well as declining diesel sales because of the diminishing price advantage of diesel fuel relative to gasoline strongly inhibit the incentive to further improve the small automotive indirect injection diesel engine. A proposed quiet, direct-injection, high-speed diesel engine with an improved (by another 15%) fuel efficiency was initiated by BMW and Steyer in the early 1980s but was shelved because of economical reasons and the technical difficulties encountered.

4. Two-Stroke, Spark-Ignition Piston Engines

Because they were forced out of the vehicular market more than ten years ago by emission legislation, the developers of two-stroke engines are

making a quiet effort to regain their place in the market. All vehicular applications-oriented research with two-stroke engines is concentrated in Japan, Switzerland, Ireland, and Australia and has produced significant results. The emission problem of two-stroke engines has been essentially eliminated by introducing cylinder fuel injection and stratified combustion techniques that allow the engine to be operated unthrottled over a wide operating range. The part-load fuel efficiencies independently demonstrated with advanced two-stroke engines in Australia, Japan, and Europe are equivalent to or better than those of an automotive diesel of current design. The pacing problem is that of a high-speed, fuel-injection system capable of reliably handling small amounts of fuel. Developers are confident that the two-stroke engine in the advanced form will make a comeback in the small-vehicle market in 3 to 5 years. Ongoing R&D work with two-stroke engines in the United States is primarily concentrated on improving the performance of carbureted outboard marine and utility engines.

5. External-Combustion Engine

R&D work with automotive Brayton and Stirling engines is concentrated in the United States but is progressing slowly because of reduced government funding. The use of high-temperature materials (ceramics) in the Brayton engine, and of high-pressure hydrogen with its associated sealing difficulties in the Stirling engine, are the pacing problems. There is little hope that any of the external combustion engines will mature by the end of this decade.

6. Free-Piston Engine

After a long period of dormancy, sporadic project work and experimentation with diesel, Otto and Stirling free-piston engines have resurfaced in Europe and the United States, using new technology and design approaches. A power output of 75 kW at 5000 cycles per minute has recently been demonstrated by Stelzer in Germany with a spark-ignited, carbureted, free-piston engine.

In conjunction with a directly coupled linear alternator, free-piston engines are believed to have potential for the generation of electrical energy over a wide range of power outputs from 3 to 700 kW because of their simplicity and capability of producing high compression ratios with low internal friction losses. If the predicted speeds can be achieved (up to 30,000 cycles per minute), a free-piston, linear alternator power unit will be very attractive for the generation of electrical energy in hybrid vehicles in terms of weight and packaging. According to predictions, free-piston engines are capable of delivering up to 30% more power at the same fuel consumption as a crankshaft engine using an identical thermodynamic cycle. However, none of the recent efforts has progressed beyond proving the validity and feasibility of the concept.

Part Two
Evaluation of Heat Engines

SECTION I

METHODOLOGY FOR HYBRID VEHICLE ENGINE EVALUATION

Hybrid vehicles use a heat engine in combination with an electrical powertrain that derives its energy from a rechargeable battery. The design objective is to reduce petroleum fuel consumption and air pollution. The possible combinations and uses of a heat engine in a hybrid vehicle (HV) are many and depend on mission objectives, drive cycle, and minimum vehicle performance requirements. The rationale for heat engine selection is, therefore, difficult to define in general terms, and each individual case must be judged separately. The scenario of hybrid-vehicle missions, powertrain arrangements, and related rationale for the selection of heat engines can be divided into four major categories:

A. PRIMARY ELECTRIC CAR WITH AUXILIARY HEAT ENGINE

The car is used primarily in the electric mode for short-range urban use while the heat engine is only used when the battery has been completely discharged. In this case, simplicity and low cost, engine reliability, low service needs, and manual starting capability have priority over fuel efficiency and emissions.

B. PRIMARY HEAT-ENGINE-DRIVEN CAR WITH AUXILIARY ELECTRIC POWERTRAIN

Hybrid-vehicle developers in Europe (Reference 1-1) envision a vehicle that is primarily heat-engine-driven. It is designed to depend on battery power only when operating in metropolitan areas where the use of heat engines may be banned in the future due to pollution constraints. The heat-engine design objectives and selection rationale for a hybrid vehicle of such a concept are essentially the same as those that govern the engine design and selection of conventional, primarily heat-engine-powered cars, i.e., compliance with minimum performance, drive-cycle fuel economy and emissions requirements, packaging advantages, weight, and noise are the primary criteria for the selection of the heat engine. In hybrids of this type, the engine is sized to provide for the power needed to meet vehicle performance requirements without the help of the electric motor.

C. HEAT-ENGINE RECHARGEABLE PRIMARY ELECTRIC CAR

For an in-series arrangement in which the heat engine has only the task of charging the battery whenever the state of charge has dropped to a tolerable minimum, the rationale for the selection of a heat engine is in essence the same as those for a mobile, standby generator, i.e, sure-start capability, low service needs, low noise, weight, and a low hourly fuel consumption at a constant speed are the primary criteria for heat-engine selection. The engine must be sized to meet maximum charge power requirements in an operating point near, or at its best, specific fuel consumption.

D. TRUE HYBRID HEAT-ENGINE/ELECTRIC CAR

Maximum petroleum fuel efficiency and versatility to adapt to various mission and vehicle performance requirements can be achieved with a parallel powertrain arrangement (Figure 1-1). A heat engine and an electric motor can act in unity or alone on the wheels of the car through a transmission. This study addresses primarily the aspects of heat-engine selection relating to the parallel powertrain concept.

Depending on the type of transmission used, the heat engine for a parallel powertrain must basically meet the requirements for cars that are primarily powered by heat engines. However, because the heat engine operates only part of the time, engine service life and power-related emission requirements necessary to meet vehicle-emission standards in terms of grams per mile are not as stringent as for cars that are powered primarily by heat engines. The control strategy usually applied uses power blending above a prescribed vehicle speed. If the vehicle starts up with the electric motor alone, the operating range to be covered by the heat engine is narrower; and the requirements for low-speed, part-load fuel economy are less demanding than those in a conventional, primarily heat-engine-powered car. Future hybrid vehicles may take advantage of a continuously variable transmission (CVT) to further improve petroleum fuel efficiency. In this case, the engine characteristics on the line of optimum fuel efficiency are of primary interest.

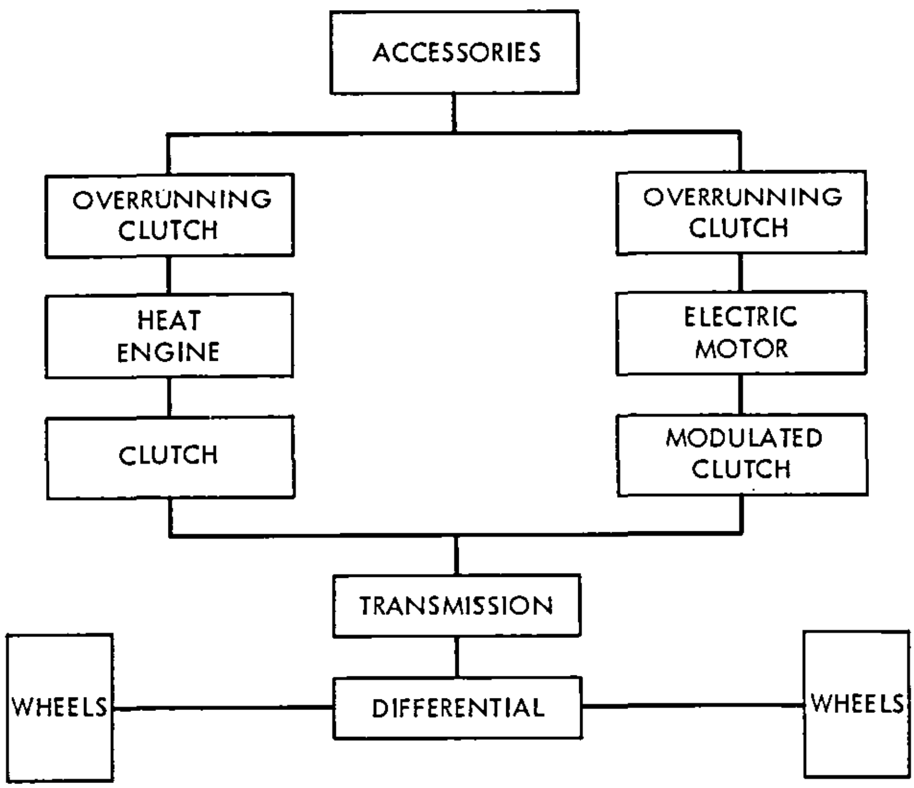


Figure 1-1. Schematic of a Hybrid Parallel Powertrain

The efficient use of the parallel powertrain concept requires that the heat engine be fired up on driver or battery demand and be shut off if drive-cycle vehicular power demands can be satisfied with the electric motor alone, and when no battery charging is required. To satisfy this requirement, the heat engine must have the capability of starting and delivering full power almost instantly. It must be compatible with a repeated stop-start operation in short time intervals (seconds) at low ambient temperatures throughout the engine service life.

To keep transient emissions to the lowest possible level, the fuel induction system of a hybrid engine unit must have the capability of starting and shutting off fuel flow instantly at the source of injection to avoid the dumping and dribbling of raw fuel into the exhaust. Practical experience (Reference 1-3) has shown that a continuous-flow (K-Jetronic-type) fuel-injection system has a definite handicap. From the standpoint of start and transient energy losses and to minimize transient emissions, it is desirable to start fuel flow no earlier than safe firing can be achieved. To accomplish this, the cranking (motoring) power requirements of the engine must be low. It is desirable to have a hot exhaust to get converters (catalytic or thermal) up to temperature and efficiency quickly. During frequent start-stop operation the catalytic converter rarely reaches the efficiency produced by a continuously operated heat engine (Figure 1-2).

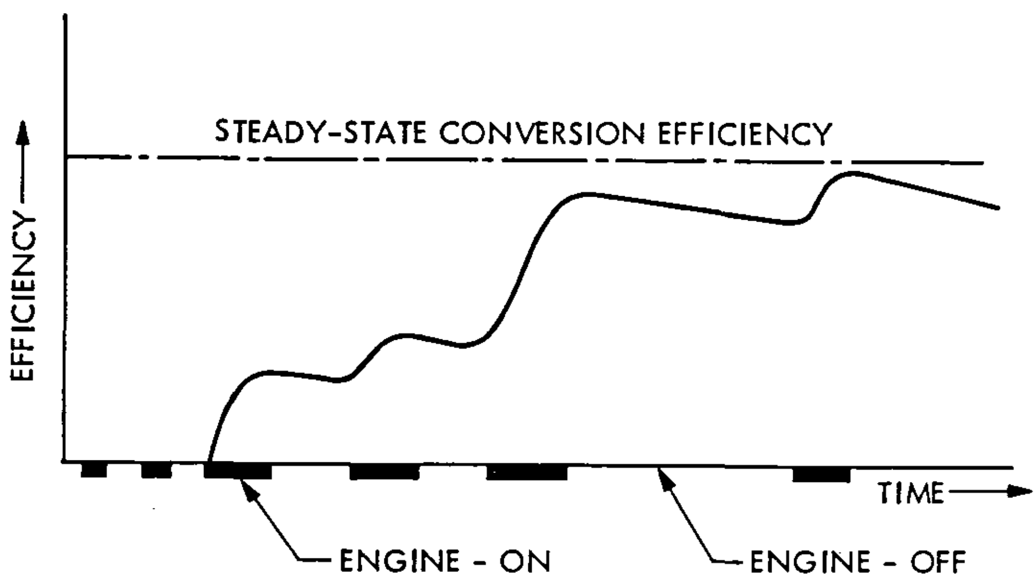


Figure 1-2. Catalyst Behavior with On/Off Engine Operations (see Reference 3)

Practical experience with the hybrid test vehicles currently tested has also shown that packaging advantages, access to the engine, and low service needs are extremely important factors to ensure that the engine performance, fuel economy, and emission criteria can be sustained throughout the vehicle service life. As can be seen from Figure 1-3, because of the size of the battery pack and the large number of required components, there is considerably less space available for the engine in a hybrid vehicle than in a primarily heat-engine-driven car of comparable performance. The possiblity of a flexible, decentralized arrangement of engine-associated systems and accessories is desirable from the overall integration and packaging standpoint.

To avoid gearing losses and to reduce dynamic interactions to a possible minimum, the engine should have a speed capability that allows for an ungeared direct-in-line drive with a motor generator or alternator. It is also desirable that the engine be sufficiently short to make a tranverse in-line arrangement of the heat engine and the motor generator possible. In current test vehicles (see Figure 1-3), the heat engine and the motor generator are parallel to each other and are inter-connected by a chain drive. A low engine inertia facilitates clutch modulation and keeps the size and weight of clutches to a minimum. According to current test experience (References 4 and 1-3), it is difficult to avoid overheating the clutches without introducing jerky shifts.

Noise is an important factor in a hybrid vehicle because of the drastic changes frequently occurring between the relatively low road and air noises dominant during electric driving and the entirely different noise spectrum suddenly produced when the engine is engaged.

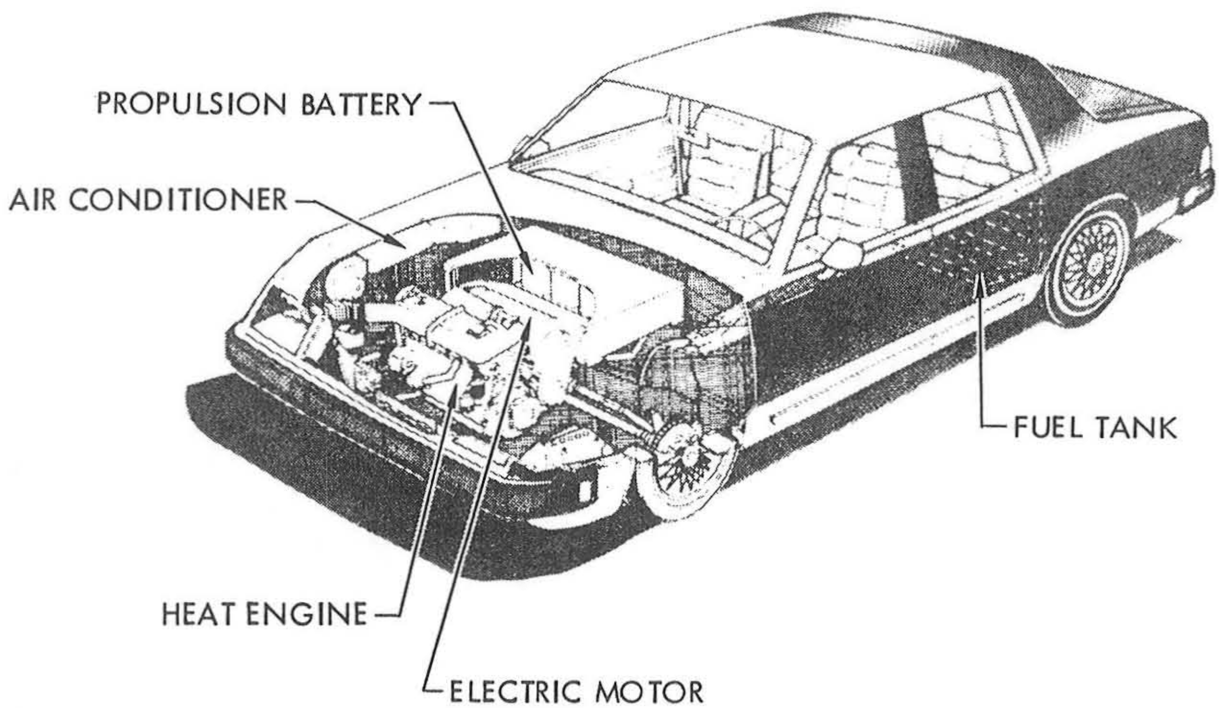


Figure 1-3. General Electric test vehicle (see Reference 4)

SECTION II

STATUS REVIEW OF VARIOUS HEAT ENGINE CONCEPTS

Engine concepts under consideration are internal and external combustion engines of advanced design that have been primarily conceived with vehicular use in mind. Only those concepts most representative for a specific concept are discussed. The major design and rating criteria for these engines are summarized in Table 2-1. The data and discussions are based on open literature and information obtained from personal communication with the indicated sources. Over 100 papers were reviewed and 19 sources were contacted for consultation. The development status, projections, potential for improvement, and pacing problems associated with concepts under consideration are discussed in the following subsections.

A. SPARK-IGNITION ENGINES

1. Four-Stroke Piston Engines

For the projected ten-year time frame the three-way, catalyst-controlled, continuous-flow electronically fuel-injected, naturally aspirated, four-stroke, spark-ignition piston engine will dominate the market for passenger cars that are primarily powered by heat engines. Multi-source fuel injection has brought an improvement of about 15% over the single-source carbureted or fuel-injected engine by significantly reducing the uneven distribution to the cylinders.

A few developers (Saab, Buick, Audi, and others) offer turbocharged, electronically knock-controlled power plants. The relative high cost for the maintenance and repair of turbocharged engines, however, inhibit the broad acceptance of turbocharging by the general public. The high speed and power boost potential of the turbocharged engine is only attractive to a small number of users. Usually there are no economical advantages although some developers claim that fuel economy advantages can be shown with wastegated turbochargers sized for maximum torque at medium engine speeds, with a reduced rear-axle ratio and an undersized, lighter engine. Although turbocharger lag has been greatly improved during recent years due to lightweight rotor design and the application of wastegate control strategy, costs and upkeep remain a deterrent for the users of a car that operates predominantly at low to medium speeds.

Besides general improvements that are currently being made in engine design, materials, components, and production technology, some major developments are being made in the improvement of the naturally aspirated, spark-ignition engine by increasing its breathing capability and compression ratio. While the potential to further improve the breathing capability obtainable with two poppet valves appears to be fairly exhausted, a sizable potential to improve partload fuel economy still exists by increasing the compression ratio. Special precautions, however, have to be taken to inhibit knocking.

Table 2-1. Comparison of Engine Design and Performance Criteria

Engine Concept	Internal Combustion Engines						External Combustion Engines	
	Naturally Aspirated Four-Stroke Spark-Ignition Engine			Naturally Aspirated Two-Stroke Spark-Ignition Engine		Diesels Indirect Injection[a] Naturaly Aspirated (Turbocharged)	Brayton	Stirling
Criteria	Piston		Rotary (Wankel)					
Engine Designation	Audi 1700	May	KKM-871	ATAC	Suzuki	Rabbit	GT-100	Mod I
Developer, Researcher	Audi/WV	Ricardo	NSU	NCAL	OEC	VW	DDA	MTI
Status	Production	R&D	R&D	Research Engine	Research Engine	Production	R&D	Research Engine
Maximum Power Output, kW/rev/min	55/5000	88/6000	121/6500	16/4500[a]	80/6000	37/5000	75/68000	54/4000
Displacement, (No. Cylinders, Rotors[b])	1.7(4)	2.0(4)	3.0(2)	0.372(1)	1.2(3)	1.5(4)	-	-
Compression Ratio	8.4	15	9.5	7.5	N/A	22	4.5 PR	17.5
Maximum Power BSFC, g/kWh	312	265	380	306	300	315	200	304
Opt. BSFC, g/kWh/% power	268/63	240/67	305/35	239/67	248/75	260/30	3200/<100	225/60
Power Concentration, kW/	32.0	44	40.4	42.7	66.3		-	-
Relative Weight to Power Ratio[a]	1.0	0.78	0.80	N/A	0.4	1.2(1.05)	1.0 [a]	3.54
Relative Volume to Power Ratio[a]	1.0	0.72	0.38	N/A	0.32	1.4(1.0)	<1.0	1.64
Speed Capability, kW x rev/min	11,978	16,812	25,517 (70,000)[a]	11,339	17,925	10,498	N/A	N/A
Remarks	Continuous Fuel Injection			Pulse Fuel Injection				
	Currently in use in experi-*mental hybrid* vehicles	Highly turbulent *combustion*	Stratified combustion Dual fuel injection Dual ignition Prediction[a]	Residual gas controlled Fuel injection Unthrottled operation	Stratified dual vortex combus-*tion* Pneumatic fuel injection	Direct injection improves BSFC *and power output* by 15%	2350°F turbine inlet temperature *2-shaft regenerated* Low NOx combustion including gears[a]	15 MPa Hydrogen *(2177 psi)*

[a]Relative to conventional low-compression-ratio, four-stroke, spark-ignition engine.
[b]Rotary (Wankel) Engine.

The most successful approach taken to date is that of highly turbulent combustion brought about by high-velocity squish flow between the piston surface and the cylinder head.

According to research completed at Volkswagen (Reference 2-1) optimum fuel economy was obtained with a compression ratio of 13:1, representing an improvement of 8 to 15% over the conventional low-compression engine. An improvement of twice that order could be demonstrated for the part-load fuel efficiency. The improvement of effective compression ratios at low engine speeds that are strongly affected by valve overlap are primarily responsible. Torque was increased up to 25% at full power and up to 13% at full torque, compared with the low-compression engine. The leaning capability of the engine also improves with increasing compression ratios.

An unresolved problem with implementing high-compression ratios and squish techniques in production engines is that of flame erosion due to high hot gas velocity flow and of piston-to-head clearance tolerances. The Volkswagen (VW) high-compression engine, for example, requires a piston-to-head clearance of 0.6 mm (0.024 in.) under hot-operating conditions to obtain optimum results with RON 92 gasoline without knocking. A disadvantage of a high compression ratio is an increase in NOx emission due to higher process temperature, especially at part load. Research engines that use high compression ratios in the range of 12:5 to 15:1 and squish techniques to inhibit knocking are under development at Ricardo in England (Reference 2-2), at Porsche and VW in Germany (References 2-1 and 2-3), and at Nissan in Japan.

Engines with a Ricardo May Fireball high compression cylinder head were offered by Jaguar at a "reduced" compression ratio of 12.5 instead of 15 as an option. Porsche claims that they will be ready to market their high-compression TOP-engine in 1984. Volkswagen is currently using a highly turbulent chamber in 1.05-liter engines of their alpha-experimental series cars. Despite some unresolved problems, it is not improbable that engines with a compression ratio of 13:1 and higher will penetrate the market on a broader basis beginning in 1985.

2. Four-Stroke Rotary Piston Engines

The outstanding advantages of four-stroke rotary engines in smoothness, weight, and size have spurred further development for special applications in which the Wankel engine has been very competitive with other candidate piston engines. A typical field in which Wankel engines are still believed to have a future, for example, is that of general aviation. The R&D work carried on during recent years at Audi/NSU in Germany and at Curtiss Wright in the United States is strongly aviation-oriented. The inherent disadvantages of the Wankel-type, four-stroke rotary engine are (1) a relatively high heat-rejection rate, (2) high production cost, and (3) high HC emissions, which have inhibited its widespread use in the automotive field.

State-of-the-art representatives of advanced Wankel engine design that has resulted from an approximately 20-year R&D effort are the research engines NSU-KKM-871 and CW-RC-75, which, as their developers claim (Reference 2-4 and

2-5), are equivalent to currently designed fuel-injected piston engines in fuel economy, emissions, and service life. The improvements have been achieved primarily by the introduction of dual ignition and two-stage, continuous-flow fuel ignition, stratified combustion, and by an improved seal design capable of producing higher compression ratios (up to 9:5) at a reduced low-speed blow-by. Housing coolant flow has been improved, rotors are now oil-cooled, and advanced metallurgical methods are used to treat housing interior surfaces for increased wear resistance.

With these measures, however, the potential of the Wankel engine for further thermodynamic improvements is essentially exhausted, which makes it difficult for the Wankel engine to compete with its high-compression piston engine contenders. According to Curtiss Wright (Reference 2-6), an attractive potential still exists for the improvement of the speed capability of the Wankel engine, which is already impressive when compared with piston engines (Figure 2-1; also see Table 2-2 for identification of data source number in Figure 2-1). An improvement of the rotary speed potential by a factor of two or more is believed possible by using gas-balanced seals pressed against the housing surface at low engine speeds that will lift off the surface at high speeds. The problem of an inherently high heat-rejection rate due to a high surface-to-volume ratio could be resolved with ceramic coatings. However, further development will be required, and results are uncertain at this time.

3. Two-Stroke Piston Engines

One of the features that drove the conventional carbureted fuel-oil mixture scavenged two-stroke engine out of the vehicular field during the last

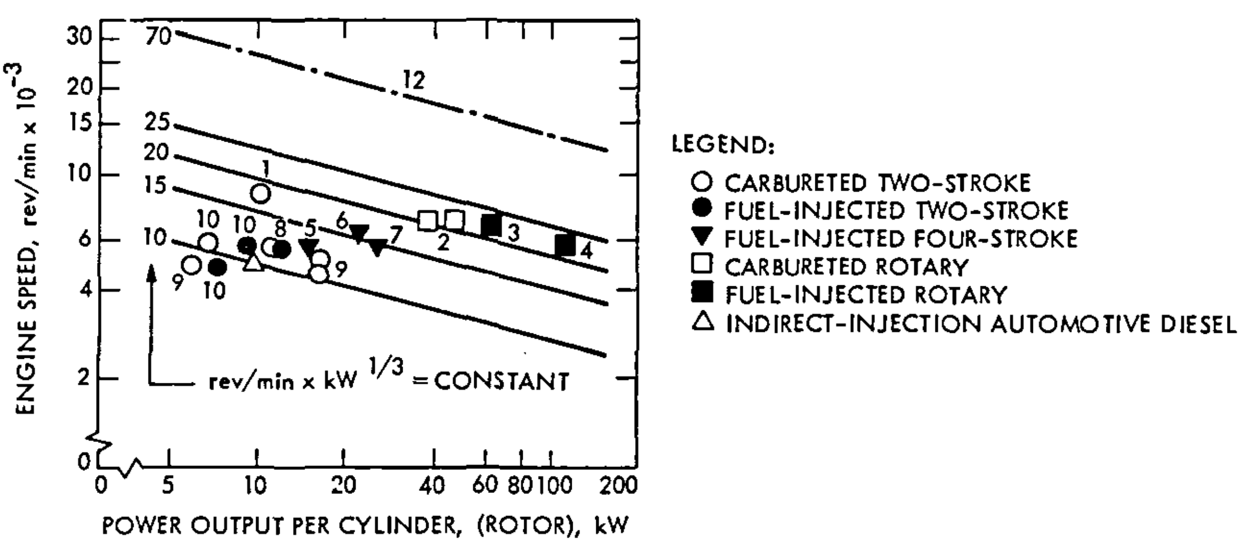


Figure 2-1. Comparison of Speed Capabilities of Engines and Engine Concepts under Consideration (see Legend Table 2-2).

Table 2-2. Identification of Data Points by Number in Figure 2-1, 2-3, 2-4

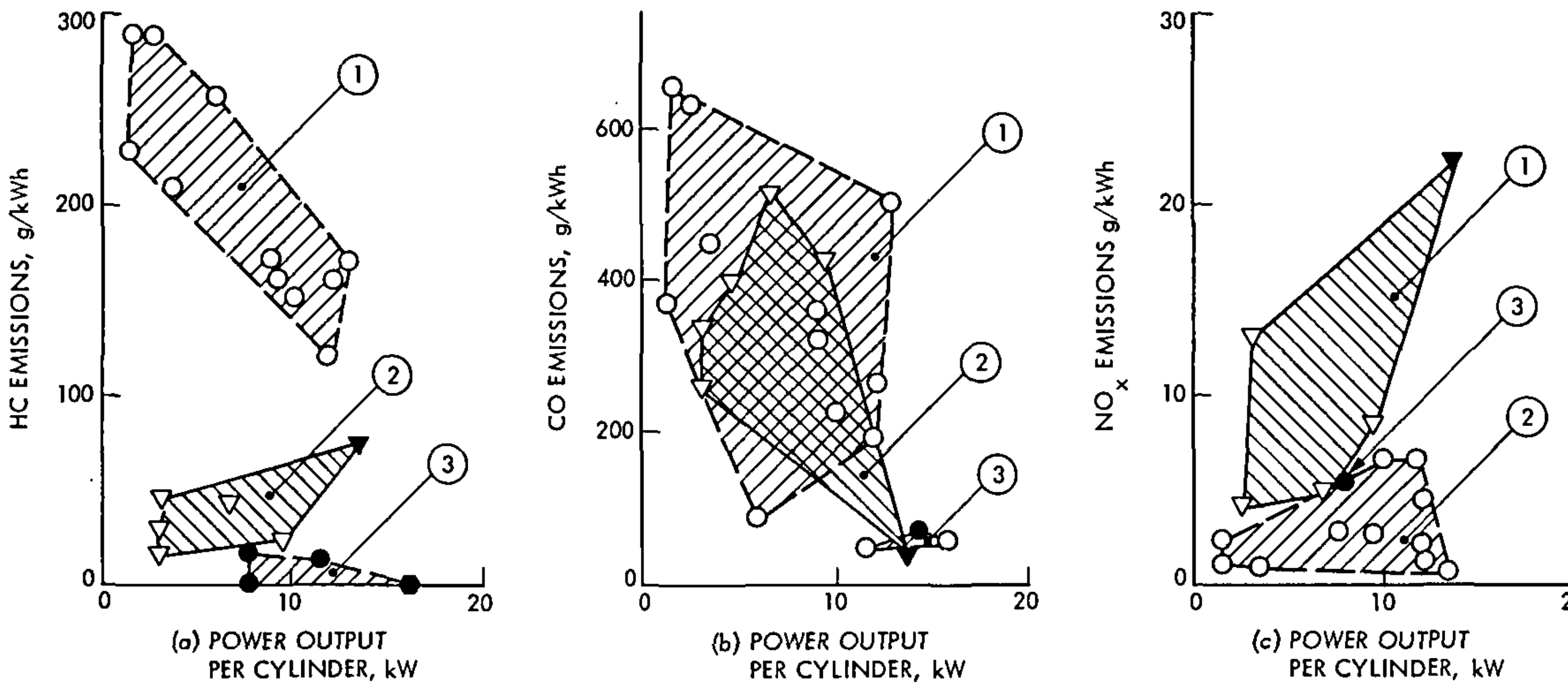
No.	Researcher or Developer	Location	Engine Type
1	Yamaha Motor Company	Japan	Racing
2	Audi NSU	West Germany	RC2-60, Production
3	Audi NSU	West Germany	KKM-877, Research
4	Curtiss Wright Corporation	United States	2-75, Research
5	Audi	West Germany	1.74-1, Production
6	May Fireball	Switzerland	Research
7	Porsche	West Germany	TOP (Thermodynamically Optimized Porsche) R&D Research
8	Swiss Instutute of Technology[a]	Switzerland	Research
9	Nippon Clean Air Lab	Japan	Research
10	Fuji Heavy Industry	Japan	Research
11	Volkswagen	West Germany	Production
12	Curtiss Wright Corporation	United States	Rotary Research with Lift-off Seals (Estimate)

[a] Eidgenoessische Technische Hochschule.

decade was an extremely high emission of carbon monoxide and unburned hydrocarbons (Figures 2-2 and 2-3) which produced smoke and an unpleasant odor. The scavenging of large amounts of fuel into the exhaust (short-circuiting) and the relatively large amounts of oil (1:20) mixed into fuel to satisfy worst-case lubrication requirements is primarily responsible for these conditions in carbureted engines. As shown in Figure 2-4(a,b), these conditions also resulted in the poor fuel economy of carbureted two-stroke engines compared to four-stroke engines. Fuel economy and hydrocarbons emissions are closely related.

These deficiencies, which have been considered "inherent" for a long time, have now been overcome as the result of research conducted in Japan, Switzerland, and Australia. Better fuel efficiencies than those for indirect injection diesels as well as emissions comparable to those for four-stroke engines have been demonstrated with research fuel-injected, two-stroke engines (see Figures 2-2 and 2-4).

According to publications and personal communications with Japanese researchers (Reference 2-7), improvements demonstrated with Japanese research engines have been achieved primarily by using cylinder fuel injection, carefully metering and minimizing the amounts of lubricating oil, and controlling cylinder charge with the amounts of residual gases in the cylinder with an



LEGEND:

- ○ CARBURETED TWO-STROKE
- ● FUEL-INJECTED TWO-STROKE
- ▽ CARBURETED FOUR-STROKE
- ▼ FUEL-INJECTED FOUR-STROKE
- ⬢ FUEL-INJECTED TWO-STROKE WITH THERMAL REACTOR

AREA 1 PRODUCTION TWO-STROKE ENGINES
2 PRODUCTION FOUR-STROKE ENGINES
3 ADVANCED RESEARCH TWO-STROKE ENGINES

Figure 2-2. Emission vs. Power Output per Cylinder

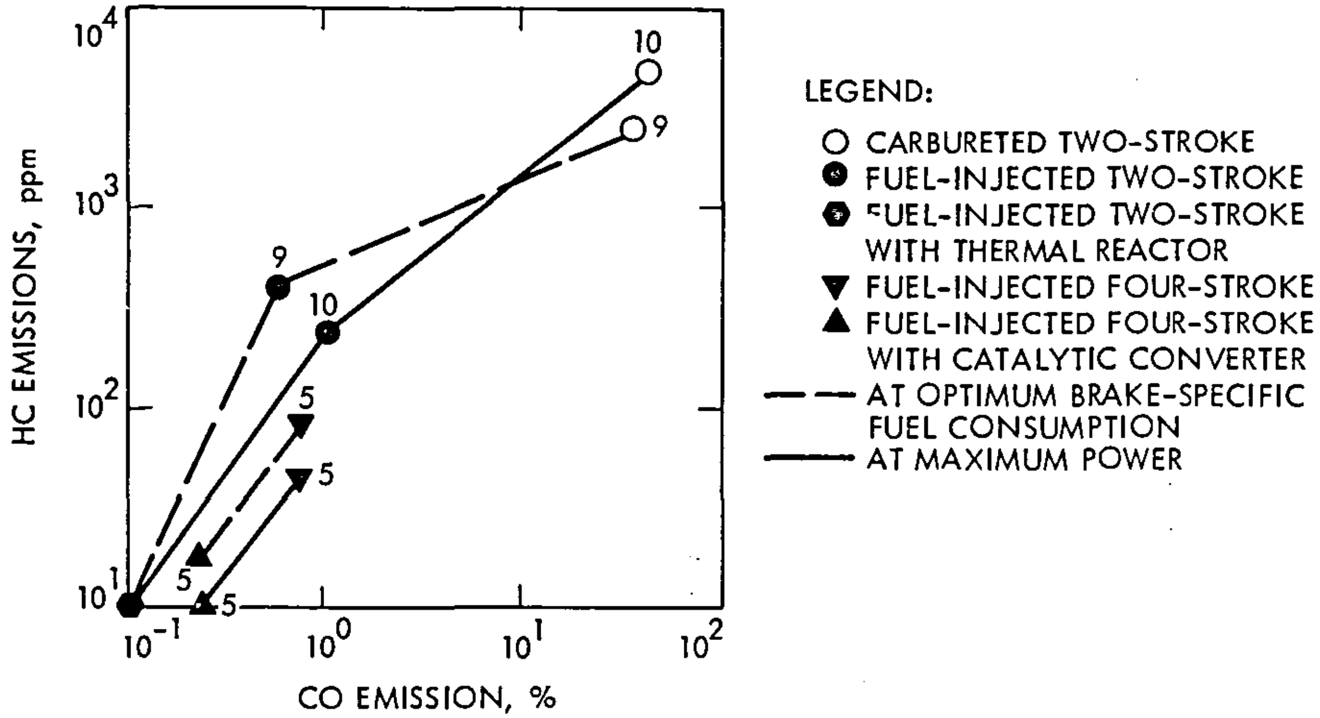


Figure 2-3. Effects of Fuel Injection and Exhaust Aftertreatment on HC and CO Emissions of Two- and Four-Stroke Engines for Optimum Brake-Specific Fuel Consumption and Maximum Power (see Table 2-2 for identification of data points by number)

exhaust throttle that permits reduction of the torque level over a wide speed range without throttling the inlet. These improvements are largely responsible for the achievement of diesel-equivalent fuel efficiencies with the spark-ignition engine. A research engine implementing this concept is the Active Thermo Atmosphere Combustion (ATAC) engine (Reference 2-8) under development at the Nippon Clean Air Laboratory in Japan. Figure 2-5 shows the performance and emission maps produced with this engine, which are still unpublished at this time.

A strong contender to the Japanese engine is a two-stroke engine under development at the Orbital Engine Company (OEC) in Australia (Reference 2-9). Figure 2-6 shows a front and side view comparing the OEC three-cylinder research engine with a four-stroke engine of comparable performance. Figures 2-7 through 2-10 show fuel efficiency and emission data generated with this

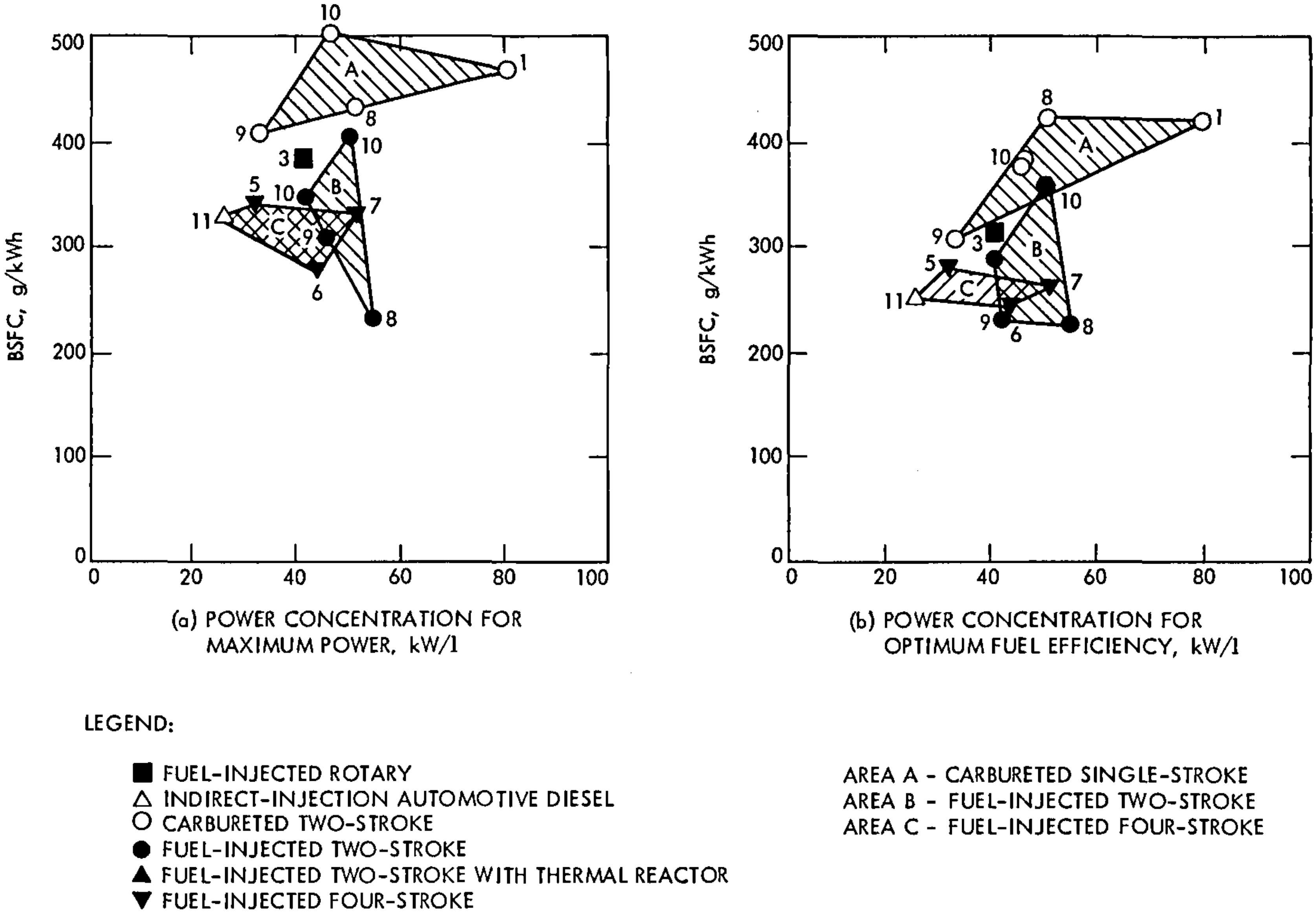


Figure 2-4. Engine Brake-Specific Fuel Consumption vs. Power Output per Liter Displacement (see Table 2-2 for identification of data points by number)

engine that have also not yet been published. As can be seen from Figure 2-11, the two-cycle engine's inherent instability at low speed is nonexistent, and the cycle-to-cycle variation achieved is comparable to those of four eight-stroke engines at lean conditions (>4). According to a presentation given by OEC at JPL on August 8, 1983, the "secret" of this extraordinary performance is a new dual-vortex stratified combustion process and a pneumatic fuel-injection system, described in Reference 2-10, that ensures proper metering and atomization of the fuel over a wide range of fuel flow rates and engine speeds. These measures together permit the operation of the engine unthrottled, controlling load primarily with the fuel-to-air ratio. As is the case with the Japanese ATAC engine, this improvement explains the achievement of a specific fuel consumption comparable to or better than that of a pre-chamber, indirect-injection, four-stroke diesel engine.

Similar results were also obtained with a cylinder fuel-injected, two-stroke engine in Switzerland (References 2-11 and 2-12). The Swiss engine uses a production Motosacoche snowmobile engine modified for fuel injection and equipped with a separate blower instead of the piston to scavenge the engine. This modification makes it possible to lubricate the engine in a conventional manner and to improve the emission quality of the two-stroke engine, which, however, does not seem to be necessary. The lube oil consumption achievable with a carefully metered two-stroke mixture lubrication system (Figure 2-12) is already as good as or better than that of a conventional four-stroke lubrication system. A definite advantage of a blower-scavenged engine, however, is its charge characteristics. As can be seen in Figure 2-13, because of the square pressure-speed relationship of the blower, the blower-scavenged engine produces an amazingly flat torque output over all its speed range. If variable in speed, the blower could be used to control the engine by controlling the scavenging ratio. On the other hand, the required blower, drive mechanism, and required external ducting contribute to unwanted bulk and significantly impair the inherent simplicity of the two-stroke engine.

The pacing problems with the cylinder fuel-injected, two-stroke engine are that of high-speed fuel injection and system reliability. The pulse frequency of two-stroke fuel injection systems is twice as high as that needed for four-stroke engines while the total fuel-injected per unit time is the same. For research purposes Swiss researchers have used a conventional four-stroke, jerk-pump system running (as with four-stroke engines) at one-half the crankshaft speed with two plungers working alternately on one cylinder. The Japanese (see Reference 2-8) have modified a conventional Bosch four-stroke injection pump to work at two-stroke crankshaft speed, which is possible under laboratory conditions for a limited time. Efforts to solve the high-speed, fuel-injection problem for two-stroke and for diesel engines are underway (see subsection II-D).

The inherent physics of the two-stroke engine is sensitive to dynamic interactions between the engine cylinder cavity and the external flow systems, the inlet, and the exhaust. The possibility of scavenging a two-stroke engine with resonance effects alone was demonstrated by Cadinnacy more than 50 years ago. Since then the exhaust system of conventional carbureted two-stroke engines have been primarily tuned to boost low end torque and to improve mid-range fuel economy by plugging the exhaust port with the returning pressure

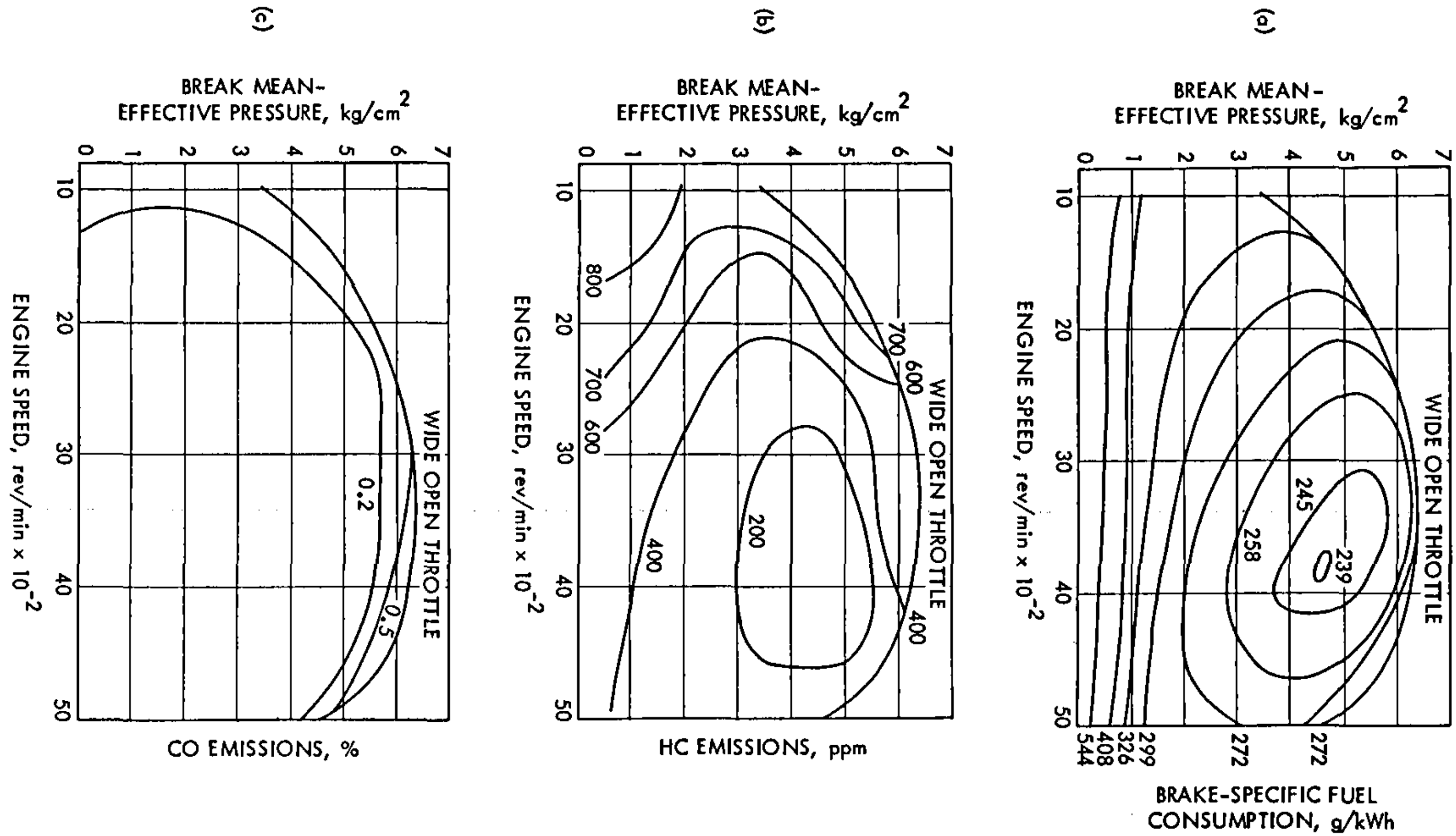
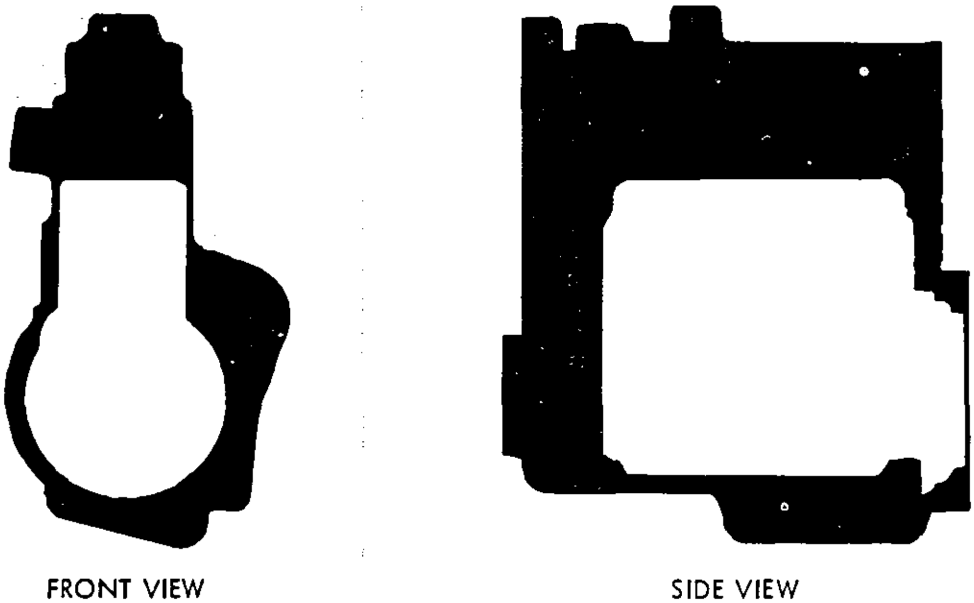


Figure 2-5. Maps Generated from Active Thermal Atmospheric Combustion Two-Stroke Research Engine (see Reference 2-7)



LEGEND:

WHITE: 1.2-ℓ THREE-CYLINDER, TWO-STROKE ENGINE

BLACK: 1.6-ℓ FOUR-CYLINDER, FOUR-STROKE ENGINE

Figure 2-6. Size Comparison of Two- and Four-Stroke, Fuel-Injected Engines of Comparable Performance (see Reference 2-9)

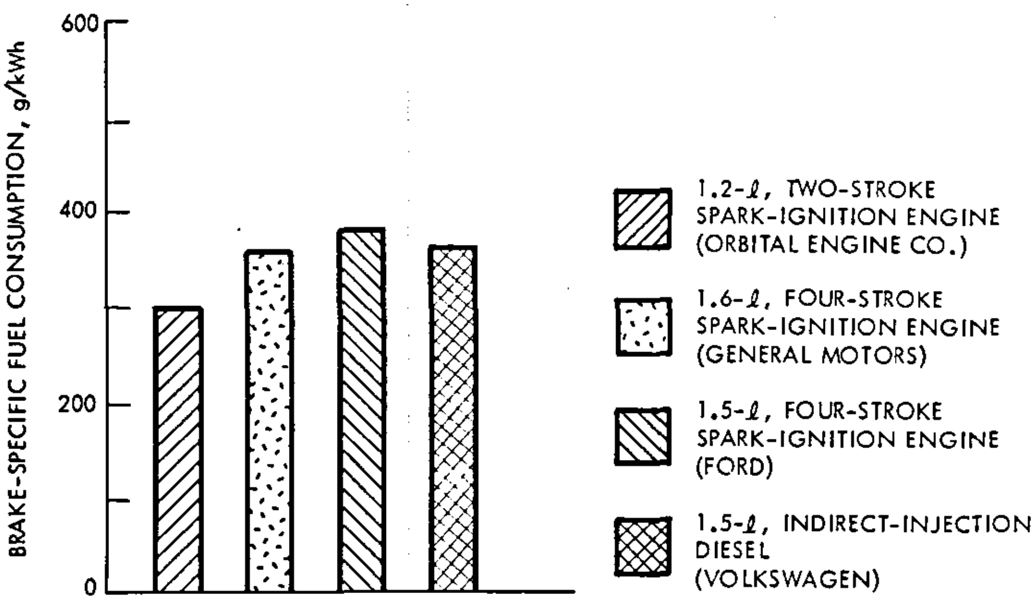


Figure 2-7. Comparison of 25% Partload Brake-Specific Fuel Consumption at 2000 rev/min (Reference 2-9)

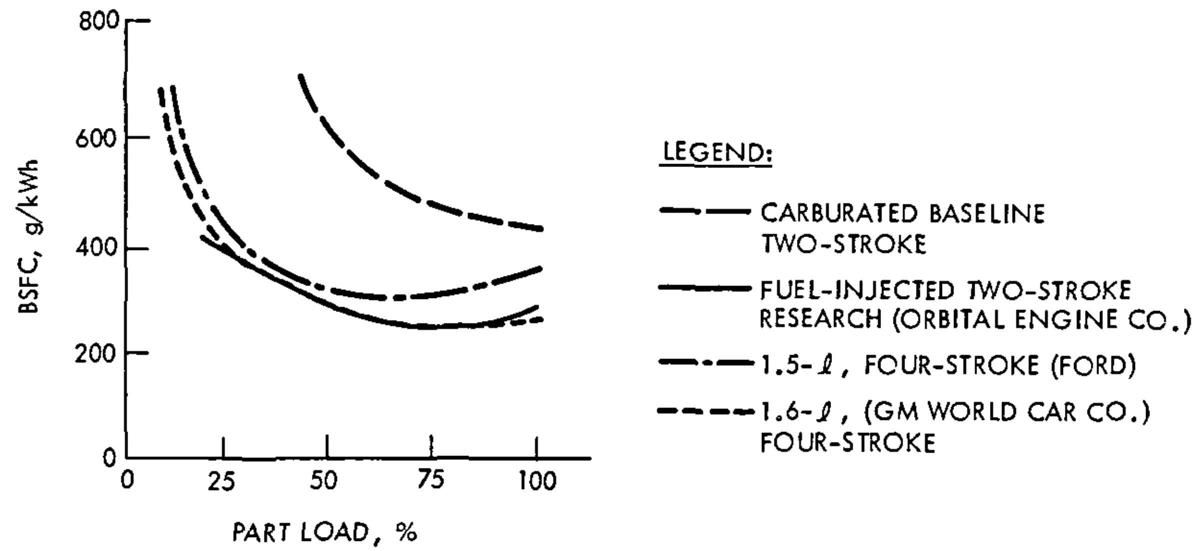


Figure 2-8. Comparison of Engine Partload Brake-Specific Fuel Consumptions in Spark-Ignition Engines (see Reference 2-10)

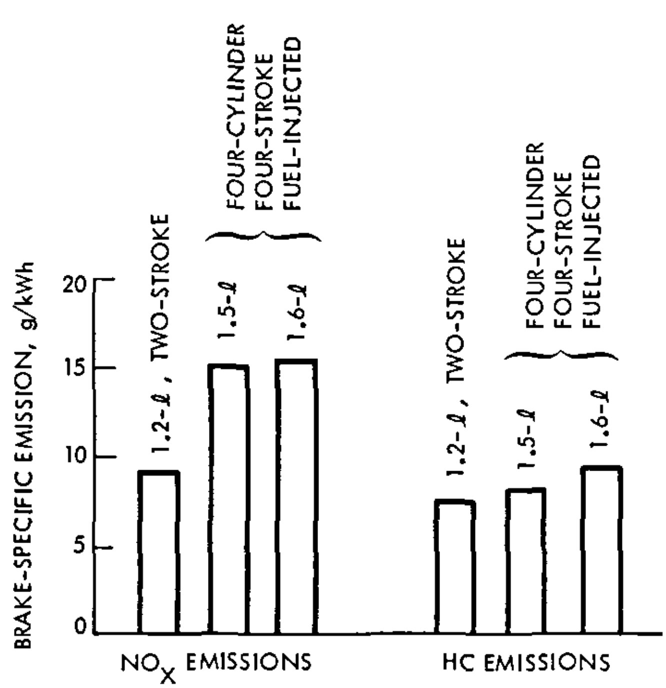


Figure 2-9. Comparison of 25% Partload Brake-Specific Exhaust Emissions in Spark-Ignition Engines at 2000 rev/min (see Reference 2-9)

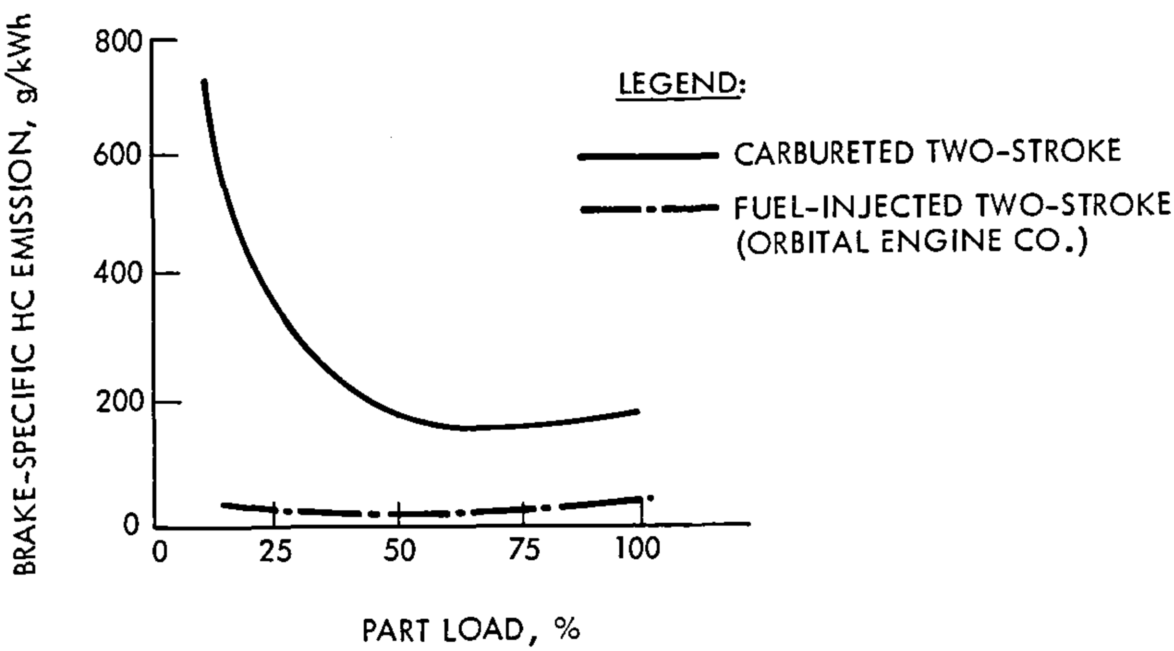


Figure 2-10. Comparison of Partload Brake-Specific HC Emissions at 2000 rev/min in Spark Ignition Engines (see Reference 2-10)

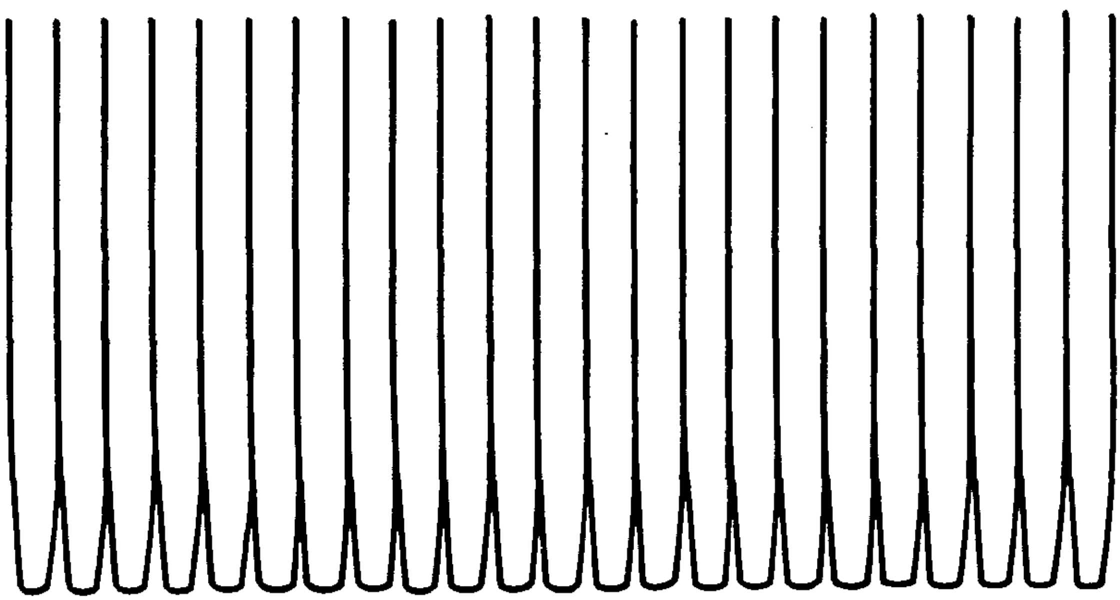


Figure 2-11. Combustion Pressure Trace of Fuel-Injected, Two-Stroke Research Engine (Orbital Engine Company, Australia) at 2000 rev/min and 67:1 air/fuel Ratio (see Reference 2-9)

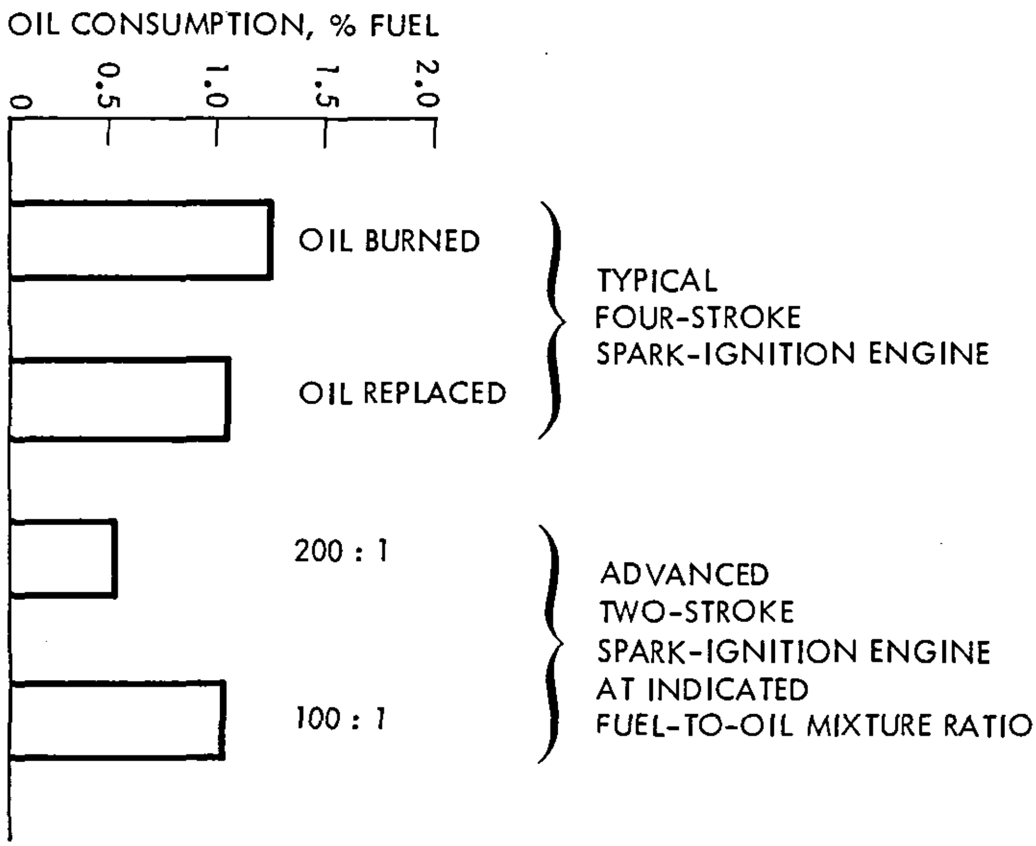


Figure 2-12. Comparison of Engine Oil Consumption Relative to Fuel Consumption

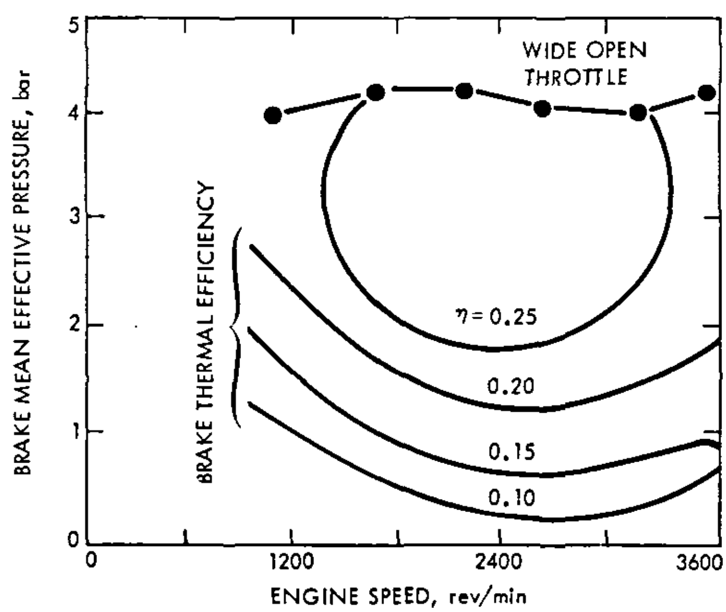


Figure 2-13. Performance Map of (Eidgenoessische Technische Hoschschule, Switzerland) Fuel-Injected, Blower-Scavenged, Two-Stroke Research Engine (see Reference 2-11)

wave as necessary to reduce short-circuiting of the scavenging flow, which was the primary cause of high HC emissions and power fuel efficiency. Broad-based systematic research and experimentation addressing the problems of exhaust tuning was conducted during the last decade at the University of Belfast (Reference 2-13). The exhaust design of most two-stroke engines in existence is based on the conclusions drawn from this research.

The tuning objectives for the cylinder fuel-injected, two-stroke engines are different from those that apply to the carbureted engine because the need to plug the exhaust port to stop short-circuiting of the scavenging flow does not exist. The fuel-injected, two-stroke engine can be tuned for maximum charge at any selected speed. The tuning rationale for an advanced two-stroke engine using the ATAC combustion concept (see Reference 2-8) may be entirely different because exhaust throttling is used to control the cylinder air-charge with the amounts of residual gases remaining in the cylinder. The improvement potential of the ATAC-engine concept with exhaust tuning has not yet been explored.

The inlet of conventional two-stroke engines is usually too short to produce noticeable resonance effects. The Yamaha Motor Company (Reference 2-14) solves this problem, for example, by providing an extra "Ton-Raum" chamber branched off from the inlet duct between the carburetor and the engine inlet (Figure 2-14). The Swiss (ETH) engine uses a resonating scavenging tube between adjacent cylinders (Figure 2-15) that boosts maximum power torque by as much as 25%.

B. COMPRESSION-IGNITION ENGINES (DIESEL)

1. Indirect-Injection Diesel Engines

Aspects of the automotive diesel engine, covering the development and market status through 1981, have been studied previously (Reference 2-15) at JPL. There have been no significant changes since that study that would change the conclusions drawn from this report.

In terms of miles per gallon the indirect-injection automotive (pre-chamber) diesel engine is, as an average, 40% more fuel-efficient than a gasoline engine of current design but only 20 to 25% if compared with base petroleum fuel. In addition, the fuel efficiency margin of the diesel narrows quickly if compared with the contending spark-ignition engine developments. Turbocharging, indirect-injection diesels have improved diesel driveability approaching the driveability expected from gasoline-powered cars. Fuel efficiency has also been slightly improved with turbocharging because of a high torque output that permits the reduction of gearing, engine speed, and friction at the same power output. With these measures, however, the improvement potential of the indirect-injecion diesel engine achievable with current technology is almost exhausted. The uncertainty of emission legislation in regard to particulates (which is still an unresolved problem), rising diesel fuel prices, and the declining sales of diesel cars have been a deterrent for producers in making large investments in future R&D work. It can be expected that no significant improvements of the indirect-injected diesel will be made

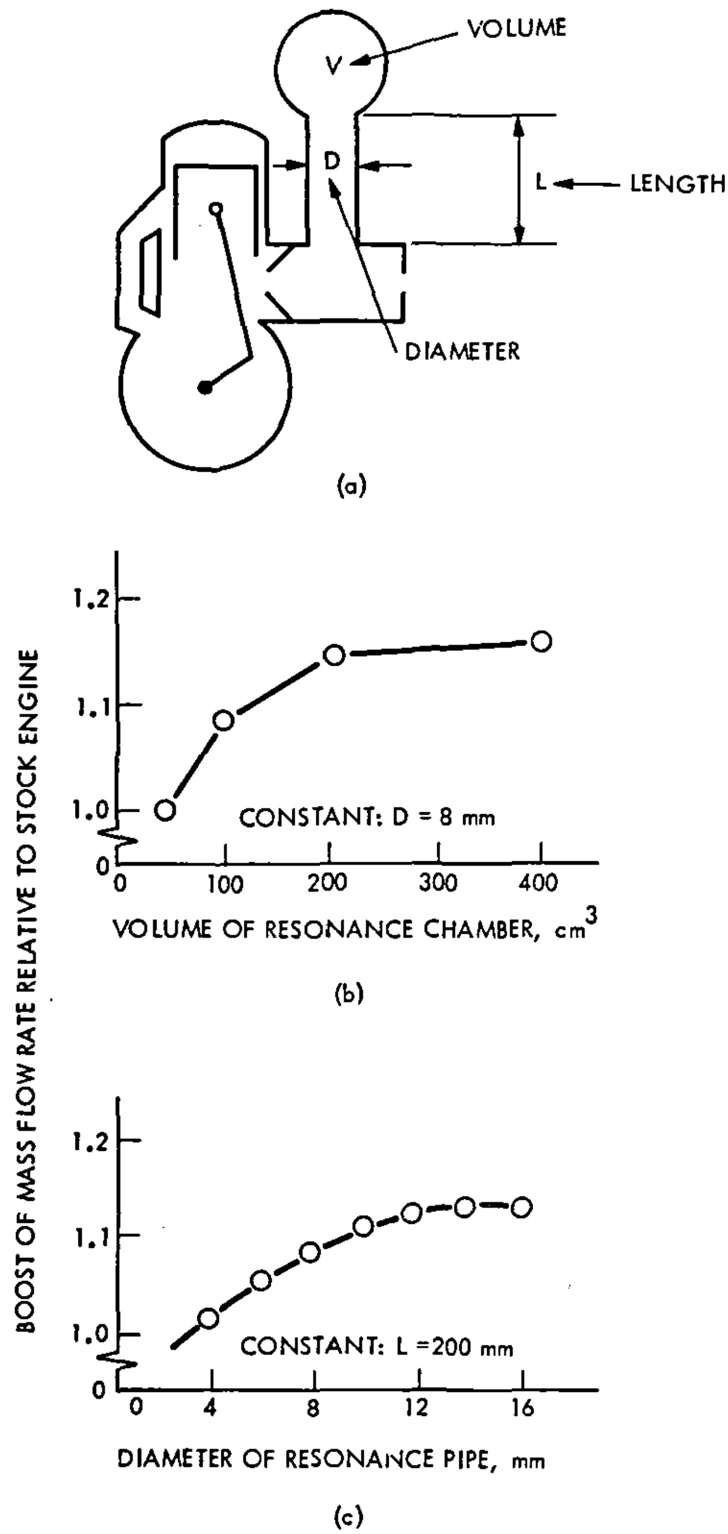


Figure 2-14. Boost Capability of Yamaha Energy Induction System at 1/4 Open Throttle and 2500 rev/min (a) System Schematic (b) Effect of Chamber Volume (c) Effect of Line Diameter, (see Reference 2-14)

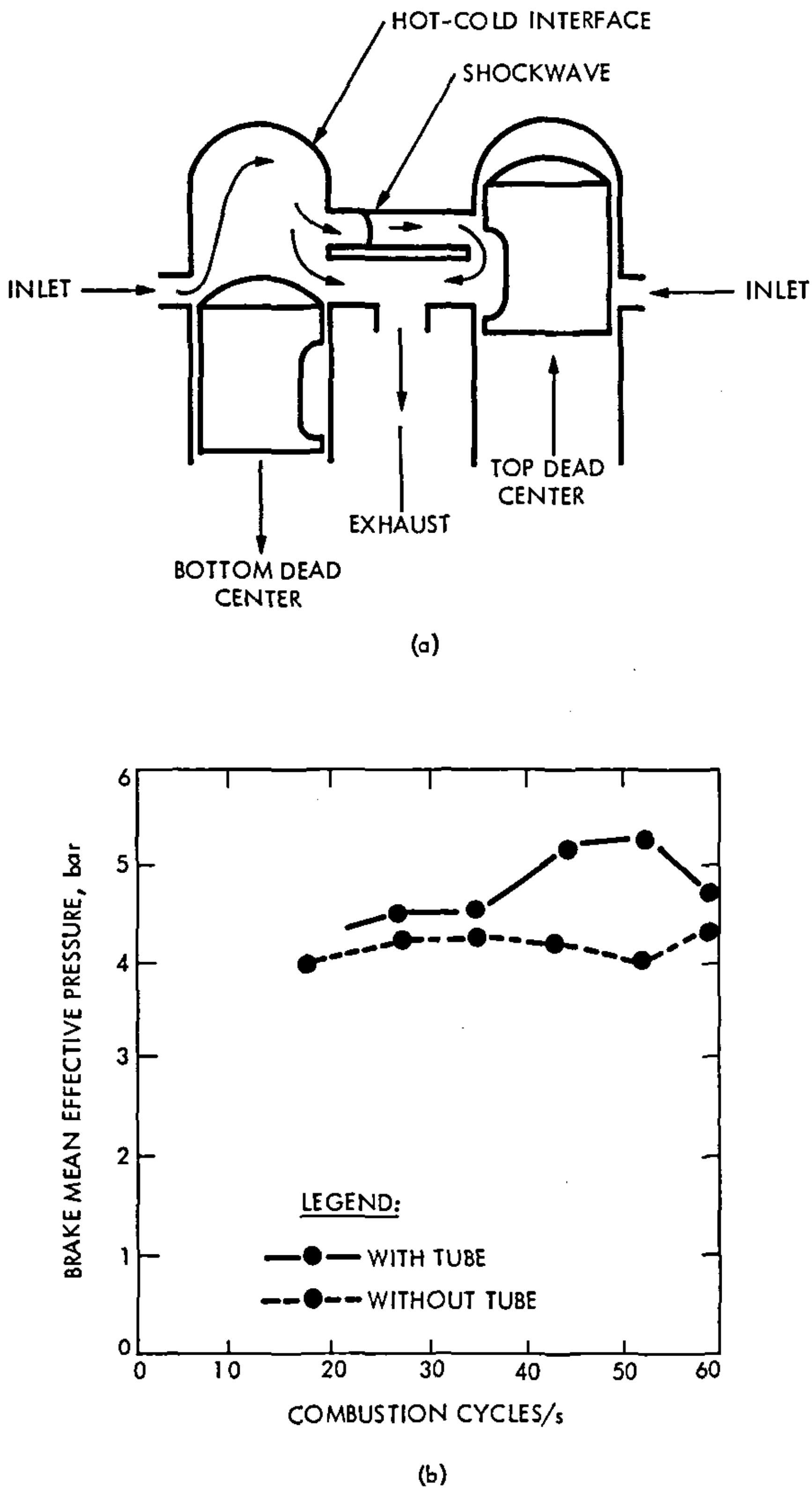


Figure 2-15. Intercylinder Resonating Scavenging Tube (ETH): (a) System Schematic (b) Boost Capability (see Reference 2-11)

within the projected ten-year time frame that will alter the conclusions drawn at this time.

2. Direct-Injection Diesel Engines

A still untapped potential for the improvement of the small, high-speed diesel is that of direct-fuel injection, which would improve its power output and fuel economy by approximately 15%. This will be necessary to maintain the lead against gasoline engine contenders of advanced design. The pacing problems are those of short-injection times, waterhammer line effects, and diesel knock due to ignition delay. In a joint R&D effort between the Bavarian Motor Works (BMW) and Austro-Steyer (Reference 2-16), the basic feasibility of a direct-injection diesel for passenger cars was demonstrated at BMW/Steyer with a design approach that provides for single-plunger injection pumps for each individual cylinder and for an accoustically attenuating engine structure. Because of the uncertain future of diesel powered passenger cars and technical difficulties encountered, this engine project was discontinued.

Efforts to improve the direct-injection diesel engine by turbocompounding and the introduction of ceramic materials are being done under U. S. Government contract at Cummins. However, these efforts cannot be expected to have a bearing on the design of passenger-car-size diesels within the projected (ten-year) time frame.

All of the R&D work on automotive diesels is exclusively oriented toward the four-stroke engine. Two-stroke, direct-injection diesels are successfully used to power trucks, railroad locomotives, and ships. There is no noticeable R&D work in progress to use the potential of the two-stroke diesel for automotive applications. As in the case of the small four-stroke diesel engines, fuel injection is the pacing problem with two-stroke, direct-injection diesel engines.

C. EXTERNAL COMBUSTION ENGINES

1. Brayton Engines (Gas Turbine)

The development of automotive gas turbines has made significant progress during the last decade in regard to component efficiency, low NO_x combustion, and the development of a usable regenerator. The inherent problems inhibiting automotive application are those of an inherently poor part-load fuel efficiency and the high operating temperatures needed to produce a competitive fuel efficiency over a wide speed and power range.

While part-load fuel efficiency can be improved to a certain extent with variable geometry, high operating temperatures from 1300 to 1400°C (2300 to 2500°F) are needed to achieve competitive fuel efficiency (Figure 2-16, Reference 2-17). The compatibility of the hot-path components with these temperatures requires the introduction of ceramic materials. The strength of structural ceramics at high temperatures has been significantly improved during the last 10 years, but their brittleness and notch sensitivity still

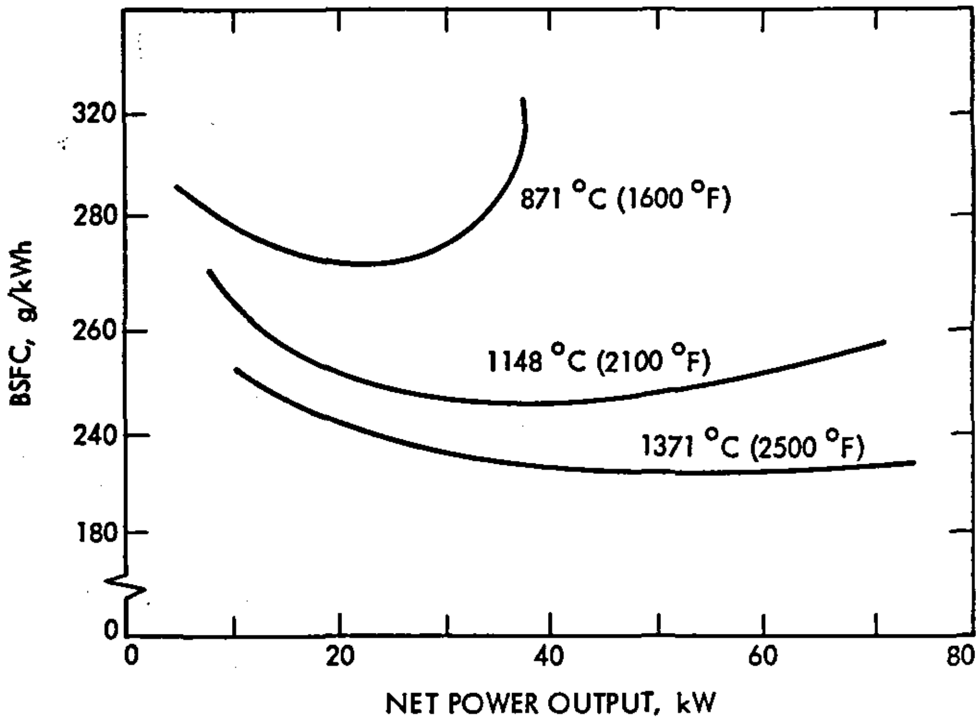


Figure 2-16. Effect of Combustion Temperature on Brake-Specific Fuel Consumption (Using Diesel No. 2 Fuel) Achievable with 75-kW (100-hp) Regenerated Automotive Gas Turbine at Standard Atmospheric Conditions (see Reference 2-17)

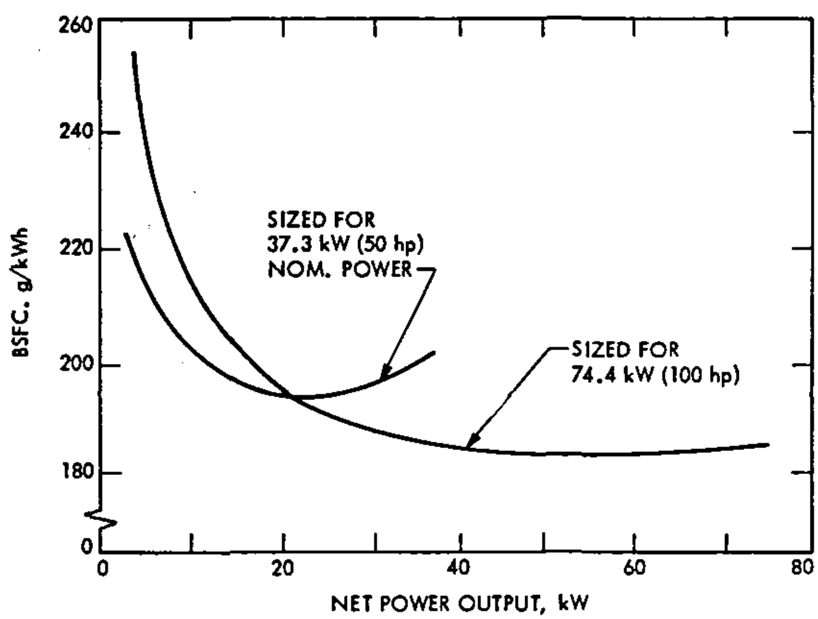


Figure 2-17. Effect of Engine Size and Power Output on Brake-Specific Fuel Consumption Achievable (at 15°C, Sea Level) with Regenerated Automotive Gas Turbine ad Standard Atmospheric Conditions (see Reference 2-19)

represent a major design problem, especially when frequent temperature changes are encountered. Differential thermal expansions of the materials make the design of unavoidable metallic-ceramic interfaces and joints problematic.

A problem also for the automotive Brayton engine is size (Figure 2-17, Reference 2-18). The efficiency of Brayton engine compressors and turbines deteriorate significantly with diminishing size due to (1) Reynolds number effects, (2) increased turbine and compressor clearance losses, (3) poor aerodynamic quality of the blading, and (4) increased leakage across the regenerator hot-cold interfaces. Unfortunately, a higher operating temperature leads also to a smaller engine for a given power output. This in turn offsets some of the thermodynamic efficiency gains achievable with a higher operating temperature.

State-of-the-art automotive gas turbines are the GT 100, under development at the Detroit Diesel Allison (DDA) Division of General Motors (Reference 2-19) and the GT 101 (see Reference 2-17), under development at the Garrett AiResearch Corporation. According to specifications of the United States Department of Energy, both turbines are designed to develop a maximum power of 75 kW (100 hp) at the driveshaft, which is considered the minimum power output producible without intolerable, small-size penalties. Using a rotary heat exchanger, both engines are basically of the same thermodynamic concept, but entirely different design approaches are taken to comply with the 100-hp requirement. With an all-concentric design and a single-shaft radial turbine, AiResearch is aiming at an optimum achievable minimum overall size and interior aerodynamic cleanliness and symmetry. With a conventionally angled-off combustion chamber feeding into a convolute and separate axial power turbine, the DDA design is more conservative and compromising for the benefit of development flexibility.

Despite optimistic projections that predicted the market penetration of automotive Brayton engines to start in 1985, it will probably take 10 to 15 years for the automotive Brayton engine to mature into a production engine. As in the case of the diesel engine, it also becomes difficult for the Brayton engine to compete with the development of advanced versions of potential candidate engines.

2. Stirling Engines

Theoretically, the Stirling engine is the most efficient heat-engine concept. As shown in Figures 2-18 and 2-19, fuel efficiencies equivalent to and better than that of an indirect injection-diesel over a wide operating range have already been demonstrated with a state-of-the-art automotive Stirling engine jointly designed and developed by Mechanical Technology, Inc, and United Stirling in Sweden (Reference 2-20).

Unfortunately, the Stirling concept has two inherent problems that inhibit its development into a competitive automotive engine. Changing the power output requires a change of the mass of the working fluid that participates in the heat cycle. For an increase in power, more mass of working fluid must be bled into the system from a high-pressure source. The other inherent problem is that of sealing. To size a Stirling engine down into a package

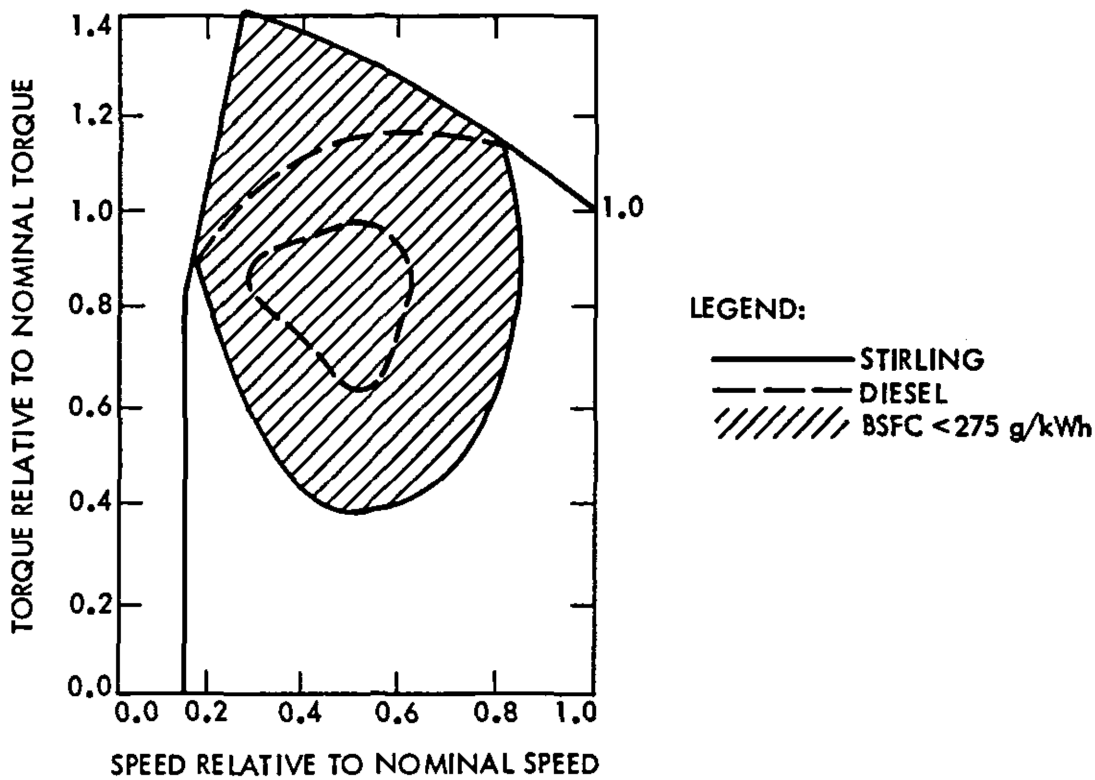


Figure 2-18. Comparison of Torque Speed Relationships of the Stirling and Indirect - Injection Automotive Diesel Engines (see Reference 2-20)

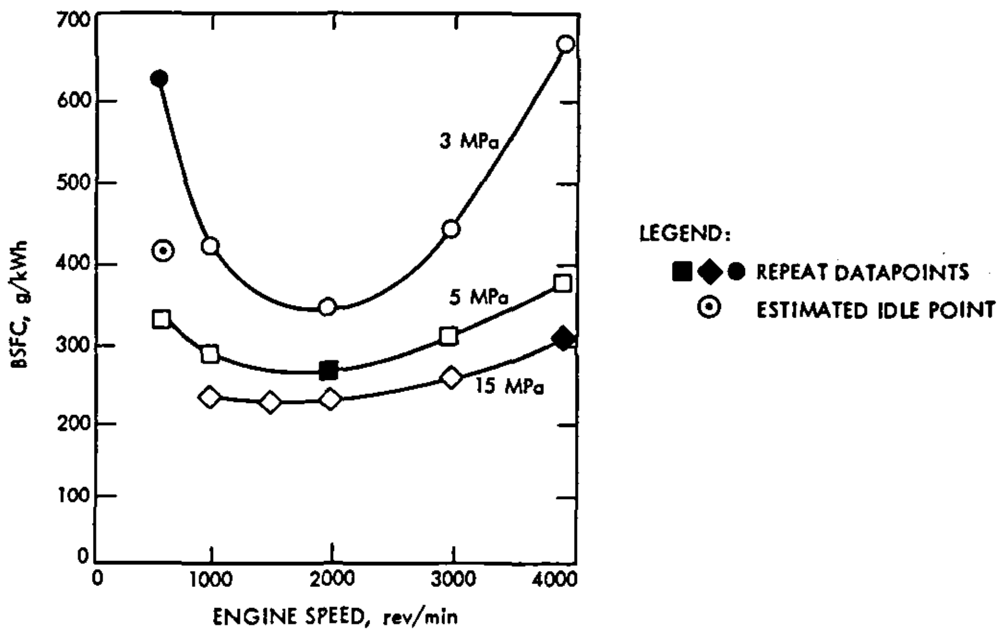


Figure 2-19. Model I Stirling Engine Brake-Specific Fuel Consumption vs. Engine Speed and Working Gas Pressure (see Reference 2-20)

that is acceptable for the designer of an automobile, hydrogen must be used as a working fluid, and high operating pressures of about 15 MPa (2175 psi) must be applied. The combination of both of these necessities imposes sealing problems that to date have not been satisfactorily resolved. As in the case of the Brayton engine, it is uncertain when the Stirling engine will mature into an automotive engine within the projected (10-year) time period.

3. Rankine Engines (Steam Engine)

The efficient use of the Rankine process requires deep expansion into the vacuum generated by a condenser, which is impractical for a passenger-car engine. For piston expanders the inherent problem is that of dead volume. To overcome this problem, the Carter automotive steam engine (Reference 2-21) uses high pressures of about 17 MPa (2500 psi) and a piston-activated poppet valve in the cylinder head, which is a concept of questionable integrity if used at high engine speeds. A variety of expander concepts have been conceived and tested but have not succeeded in producing sufficiently deep expansion without unwanted complexity and bulk.

To achieve fuel efficiencies that only approximate those of conventional spark-ignition engines, the Carter engine also applied superheating to temperatures up to 540°C (1000°F). These high temperatures impose cylinder lubrication problems that have not been resolved. Lear had the same problems with his automotive steam-engine project. Because the automotive steam-Rankine engine offers no potential for competition with current automotive designs, government supported R&D was discontinued several years ago.

D. FUEL-INJECTION SYSTEMS

Four-stroke, spark-ignition engines of state-of-the-art design are usually equipped with a single- or multi-source, continuous-flow, fuel-injection system. The fuel flow is primarily controlled by an air-flow meter in conjunction with an oxygen probe positioned in the exhaust. Engines that depend on a cyclic injection of fuel into the cylinder are (1) diesels; (2) advanced, two-stroke, spark-ignition engines; and (3) programmed combustion (Ford-PROCO), four-stroke, spark-ignition engines.

For small automotive engines, the pacing problems with cyclic cylinder fuel injection are (1) the short time available for injection, (2) the small amount of fuel to be injected per cycle, and (3) the achievement of a good fuel atomization over a wide range of engine speeds and fuel flow rates. The high-speed, fuel-injection problem becomes especially pronounced in two-stroke engines because the number of pulses to be delivered per unit time is twice as high, whereas the time available for injection and the amounts of fuel to be injected per cycle are only one half that of a four-stroke engine.

The research with two-stroke engines in Japan and Switzerland (see References 2-8 and 2-11), featured modified production, four-stroke, fuel-injection systems of the Bosch jerk-pump type. These were adequate for laboratory tests but were limited in the time engines could be operated at full power and speed. Both of the indicated researchers agree that new solutions to the two-stroke, fuel-injection problem must be found.

Reportedly (see Reference 2-12), the Federal Swiss Institute of Technology (ETH) is developing an electronically controlled cylinder, fuel-injection system applicable to diesels and spark-ignition engines that is capable of producing the typical programmed injections at high speeds (Figure 2-20). The system is said to be capable of producing 100+ pulses per second at pressures up to 600 atm. The system is an ETH proprietary item that has not been published.

The Australian researchers have taken an approach to the high-speed, fuel-injection problems of their own (described in detail in Reference 2-10). As shown in Figure 2-21, the concept uses a combination of compressed air and plunger-displacement metering to inject an extemely rich fuel-air mixture into the cylinder. This system has been successfully used in the OEC-research, two-stroke engine described in subsection A-3. It produced an extremely wide range of total air-to-fuel ratios so that partload could be controlled by means of air-to-fuel variation alone without throttling the engine.

According to press publications such as WARDS Engine Update and Automotive News, several prominent producers of fuel-injection systems such as Bosch and Lucas are making progress in the development of electronically controlled, cylinder fuel-injection systems for high-speed diesels. It can, therefore, be reasonably assumed that the fuel-injection problems of the two-stroke engine can be resolved in the future.

E. FREE-PISTON ENGINES

Because force actions are primarily of an inertial, dynamic nature, free-piston engines have a particularly attractive potential for diesels and for alcohol-fueled, spark-ignition, combustion cycles. There is no crankshaft, and high compression ratios can be produced without combustion-pressure-related friction penalties. Figure 2-22 compares the predicted friction losses and the power output of a resonating free-piston diesel engine against data obtained with a conventional two-stroke crankshaft diesel. As can be seen, the free-piston friction losses are nearly constant whereas the friction losses of the crankshaft engine rise steeply with speed by a factor of more than six. This translates into a power and fuel efficiency gain of up to 40%. The problem is how to efficiently extract the power generated by a free-piston engine.

Uniflow-scavenged, two-stroke, diesel free-piston engines of the opposed piston type were first developed by Junkers in Germany and Pescara in France between 1930 and 1950. Because of their compactness, narrow width, and smooth vibration-free operation, diesel free-piston compressors were for many years the preferred source of energy to power jackhammers on construction sites and to pressurize torpedo systems on submarines. Pescara experimentally used two or more free-piston diesel compressors in parallel as a gasifier for gas turbines to power trucks and ships. The Pescara free-piston engine (Figure 2-23, Reference 2-22) had a power output of 90 kW at 1800 cycles per minute and (including power turbine and gears) a weight of 278 kg/kW, which is light for a diesel engine. However, including the turbine and reduction gears, the Pescara power system was not competitive with diesels of conventional crank-

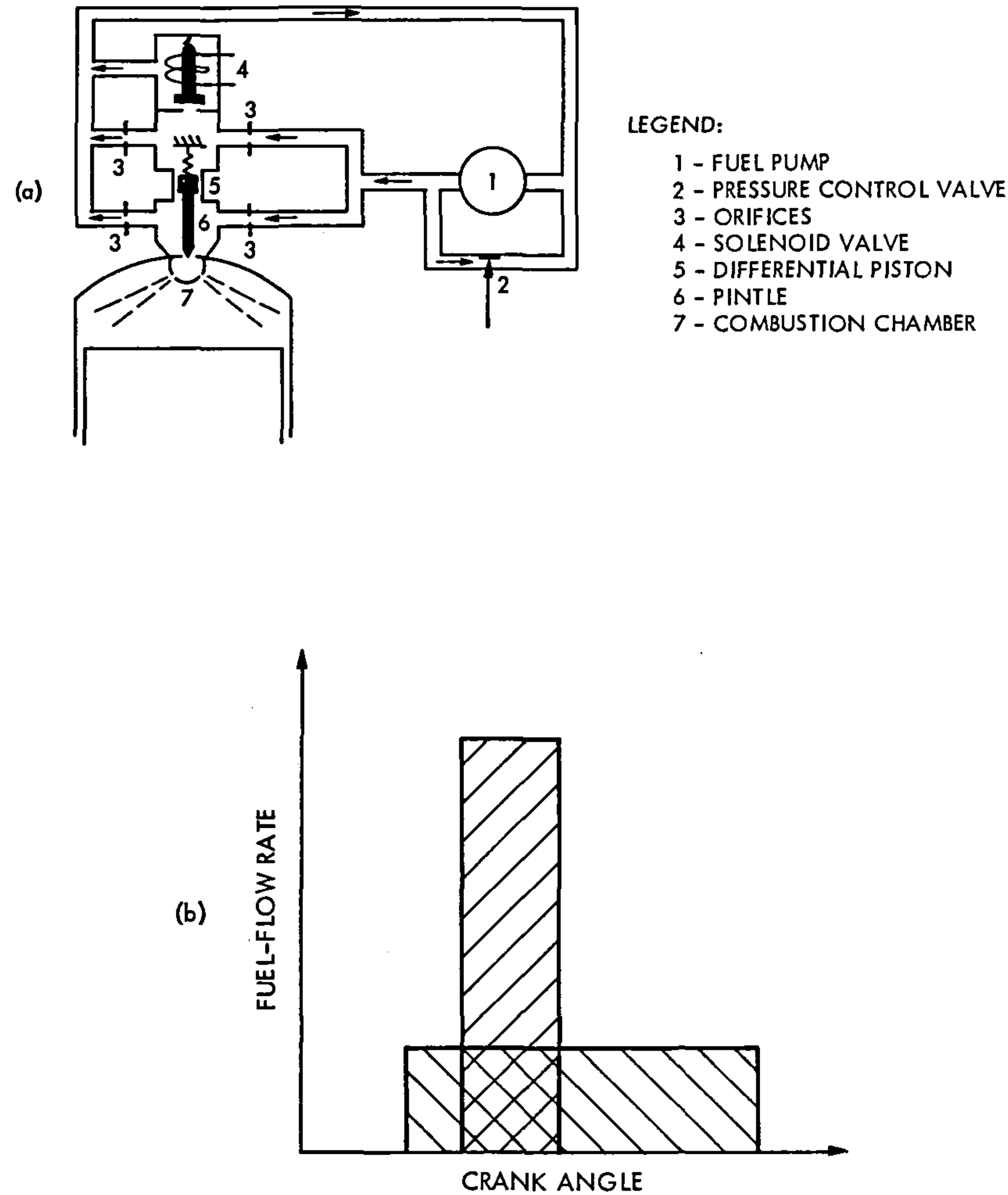


Figure 2-20. Swiss Institute of Technology Fuel-Injection System (a) System Schematic (b) Typical Fuel-Injection Pattern Characteristics Producible with the System (see Reference 2-12)

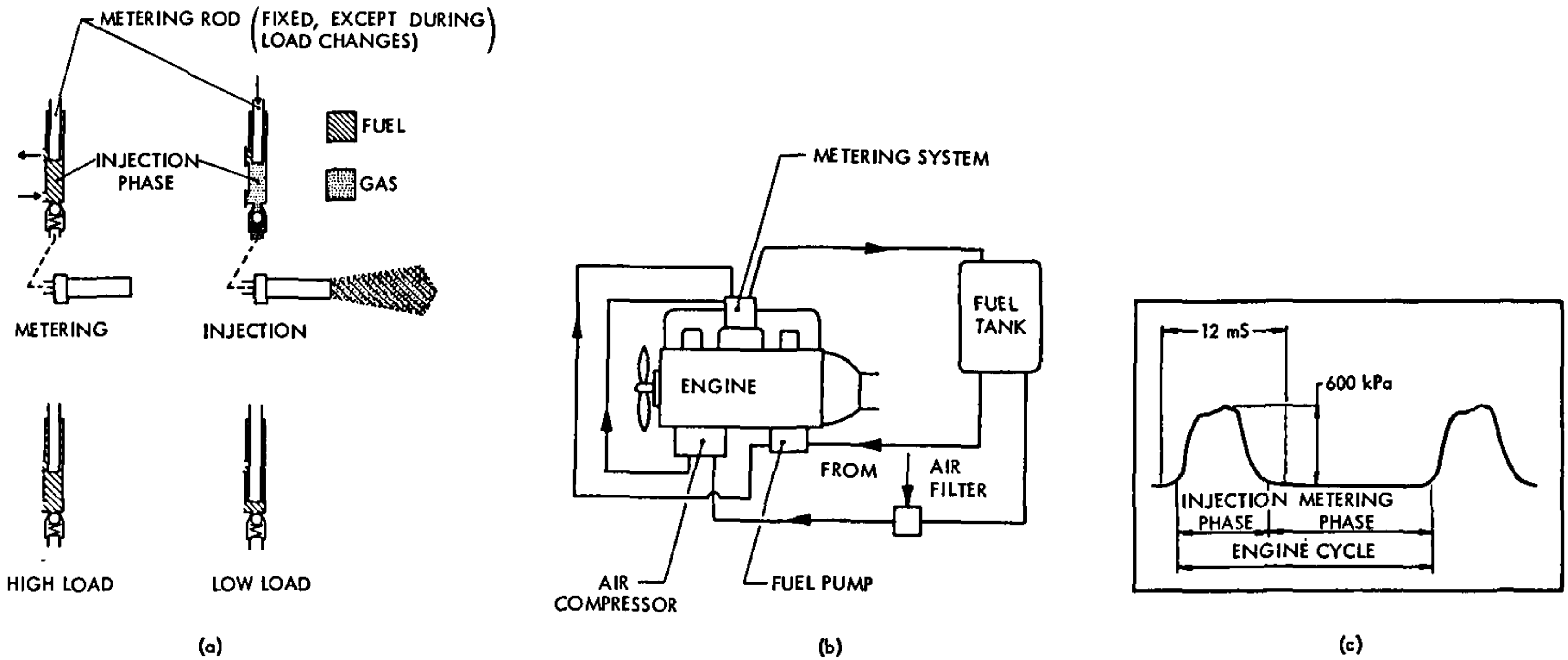


Figure 2-21. Orbital Engine Company (Australia) Pneumatic Fuel-Injection System (a) Fundamental Metering Principle (b) System Schematic (c) Relationship Metering Phase and Injection Pulse-to-Engine Cycle (see Reference 2-10)

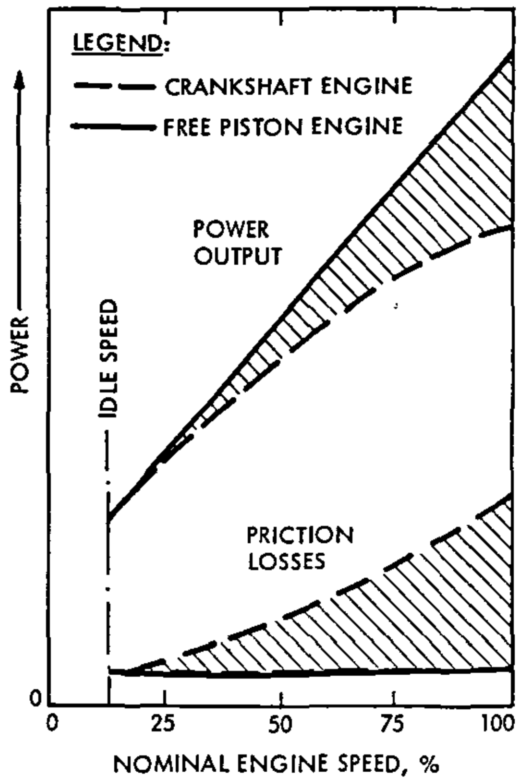


Figure 2-22. Comparison of Predicted Power Output and Friction Losses of Uniflow Scavenged, Free-Piston Diesel Engine with GM-type Blower Scavenged Crankshaft Two-Stroke Diesel Engine (see Reference 22)

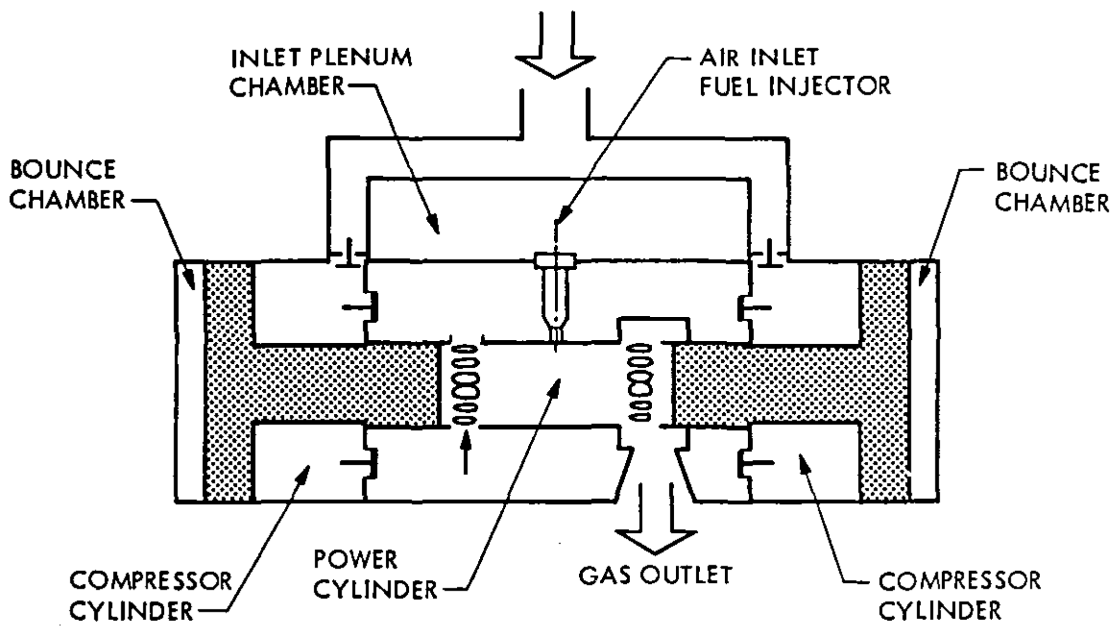


Figure 2-23. Schematic of Uniflow Scavenged Diesel Compressor/Gasifier of Conventional Junker/Pescara Design (see Reference 2-22)

shaft design in weight, bulk, and fuel efficiency. The system exhibited an optimum specific fuel consumption as high as 335 g/kWh because of the poor partload efficiency of the turbine and reduction gear losses. The approach using free-piston engines to generate shaft power was therefore abandoned.

The idea of integrating a free-piston engine with a linear alternator is an old one, but it was never realized because of the limited speed potential of free-piston engines of conventional design. The free-piston engines discussed above developed a maximum speed of approximately 1800 cycles per minute. The minimum possible weight of the oscillating masses, the dynamics of the synchronizing mechanism, and the capability of the mechanical fuel injection systems used were the limiting factors. Some research work with turbocharged spark-ignited, gasoline-fueled, high-speed, free-piston engines to generate electrical power on an aircraft was in progress in Germany during World War II. These efforts were discontinued because of difficulties encountered with piston and alternator cooling and because of other war-related priorities. However, to the best knowledge of the author, the feasibility of such a concept was successfully demonstrated.

After three decades of dormancy, sporadic research and project work with free-piston engines has reappeared in various places in the belief that the inherent problems of the past can now be overcome with advanced materials and technology. According to recent studies (References 2-23 and 2-24) linear-alternator, free-piston engine combinations have a potential for the generation of electrical power over a wide range (between 3 and 700 kW). A power output of 75 kW at 5000 cycles per minute has recently been demonstrated with a new spark-ignited, two-stage compression free-piston engine schematically shown in Figure 2-24. The developer, Stelzer, believes that the concept has the potential for oscillating speeds up to 30,000 cycles per minute. Resonance Motors in Irvine, California (see Reference 2-25), is currently studying the feasibility of a 668-cc, free-piston engine capable of oscillating at 42 to 47 Hz and developing a power output of 28.6 kW. The engine projected is shown schematically in Figure 2-25. Resonance Motors does not reveal in their report how the engine is scavenged and was not available for questioning.

The use of a freely oscillating Stirling engine without a crankshaft in conjunction with a linear alternator has been studied in concept by Mechanical Technology, Inc., under government contract (Reference 2-26). It eliminates the external hydrogen leakage problem with dynamic seals and results in a compact configuration that is more tolerable from the packaging standpoint than a crankshaft Stirling engine. The results of a recent project study in this direction are presented in Figure 2-26 and include the following specifications:

(1)	Bore	76.2 mm
(2)	Stroke	66.7 mm
(3)	Displacement	0.3 l
(4)	Compression Ratio	17:1
(5)	Engine Power Output	5.4 kW
(6)	Engine-Specific Fuel Consumption with Diesel No. 2 Fuel	310 g/kWh
(7)	Alternator Power Output	4.5 kW
(8)	Unit Overall Volume	36.3 l
(9)	Unit Weight	59 kg

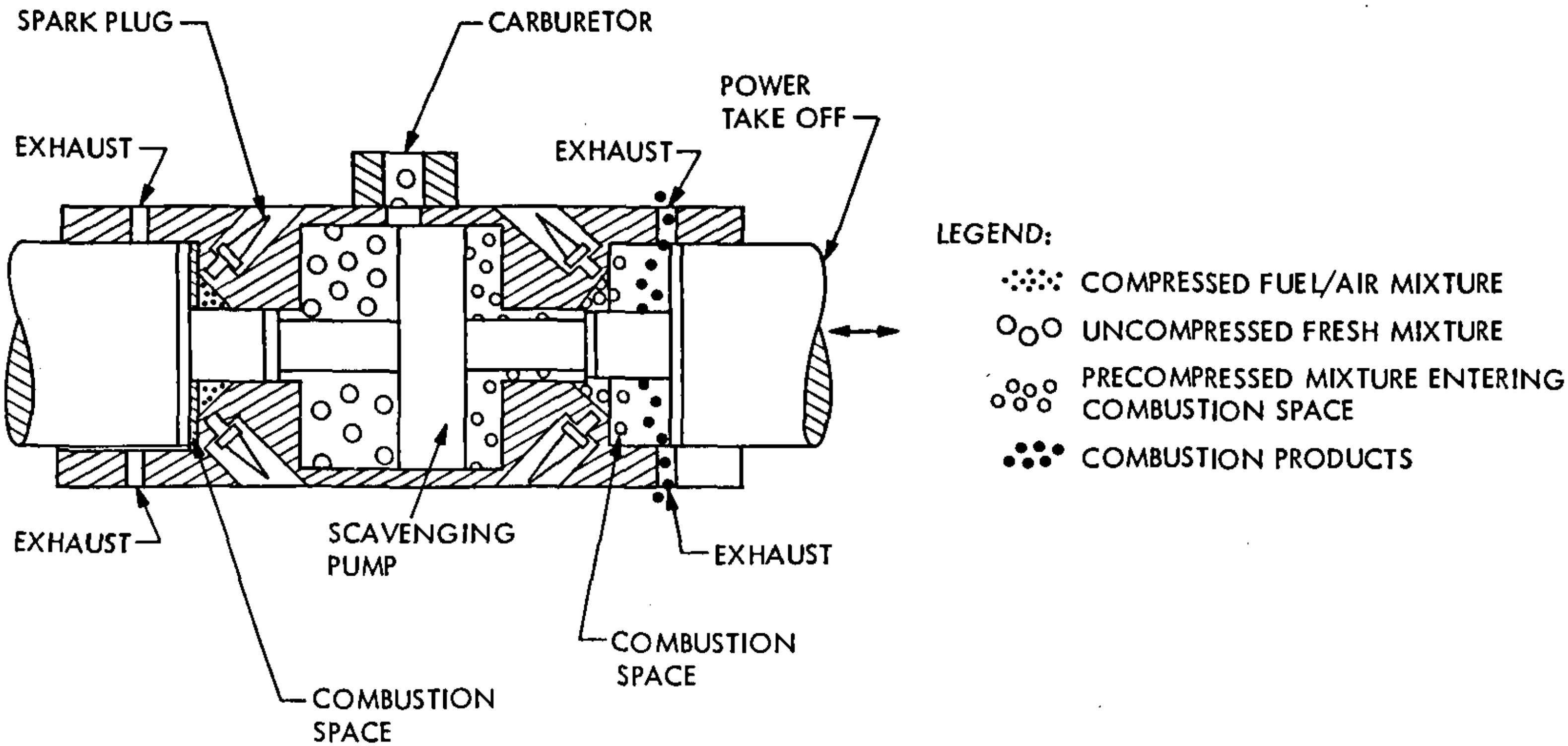
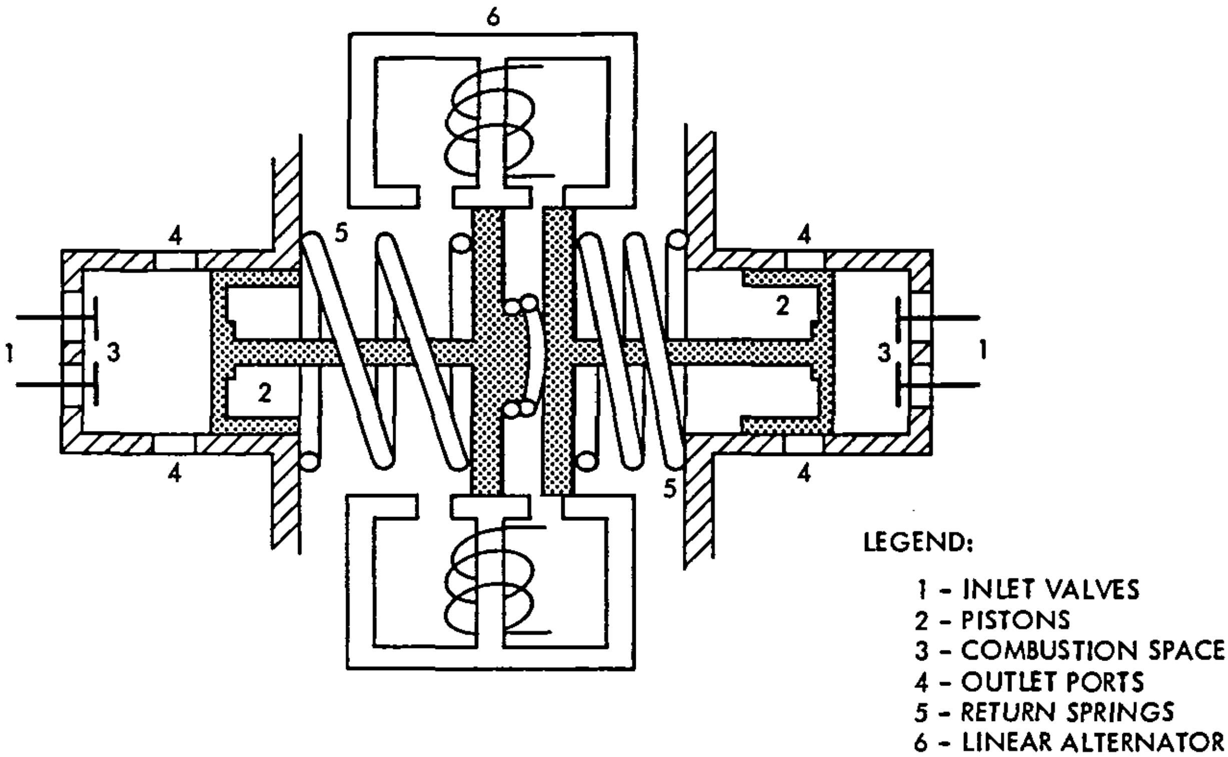


Figure 2-24. Schematic of Stelzer Free-Piston, Spark-Ignition Engine (References 2-23 and 2-24)



2-25. Schematic of Free-Piston Linear Alternator Unit by Resonance Motors, Inc. (modified from Reference 2-25)

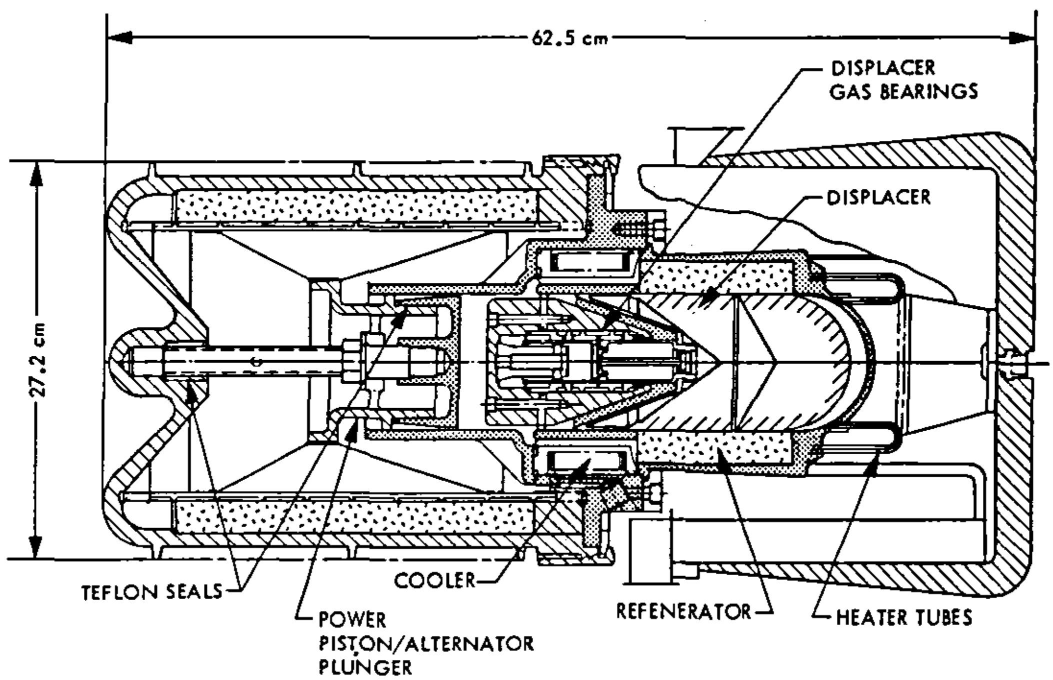


Figure 2-26. Sealed Stirling Free-Piston Power Unit (see Reference 2-26)

SECTION III

POTENTIAL FOR HYBRID VEHICLE APPLICATION

Figure 3-1 generically compares the key properties of the engine concepts under consideration. These concepts are discussed in the following subsections.

A. BRAYTON ENGINE

Fuel efficiencies that theoretically can be obtained with a regenerated Brayton engine (see Figures 2-16 and 3-1a) are very competitive, provided the projected component efficiencies and operating temperatures can be achieved with production hardware. As shown in Figure 3-1b there are no striking size or packaging advantages of the Brayton engine over its piston-engine counterparts because of the bulkiness of the heat exchanger, associated ducts, drive mechanisms, and the housings required for gearing down from the turbine to the output power shaft.

Because of its multi-fuel capability, the Brayton engine would be the ideal heat engine for a hybrid, but the gas turbine does not lend itself to hybrid operations that require rapid stop-start capability. It takes a minimum of 5 s to crank the compressor into the boot-strapping range and another 5 s or more to accelerate the rotor, gears, and the heat exchanger mass to full maximum speed. It also takes a certain time to achieve the hardware thermal equilibrium necessary to obtain good fuel efficiencies typical of a regenerated Brayton engine. Special precautions to obtain an acceptable stop-start capability could be taken but still remain undeveloped. The compressor inlet, for example, could be shut off during cranking and opened instantly upon reaching boot-strapping speed to reduce compressor and turbine work and to prevent heat exchanger coolout. Low-inertia shutters that can be operated quickly would probably be needed across the inlet and outlet plane of the heat exchanger to prevent undesirable heat losses during engine shutdowns.

The Brayton engine would have a potential for battery charging if scaled to the desired power output (20 to 30 kW). Because of small-size effects this would not be possible without strong fuel-efficiency penalties (see Figure 2-17). A direct power take-off from the turbine rotor in the form of electrical energy would be the ideal way to charge the battery of a hybrid vehicle. Assuming that a down-scaling at constant Mach-numbers without an impairment of the engine thermodynamics is possible in the future, with single-shaft Brayton engines of advanced design, for the power output of interest (20 to 50 kW), the shaft speeds would be in the order of 100,000 to 165,000 rev/min.

Unfortunately, the low level of funding currently provided for the development of automotive gas turbines, as well as the dependence on ceramic technology, makes it difficult to predict at what time an economical gas turbine might be commercially available. It seems probable that no mature Brayton

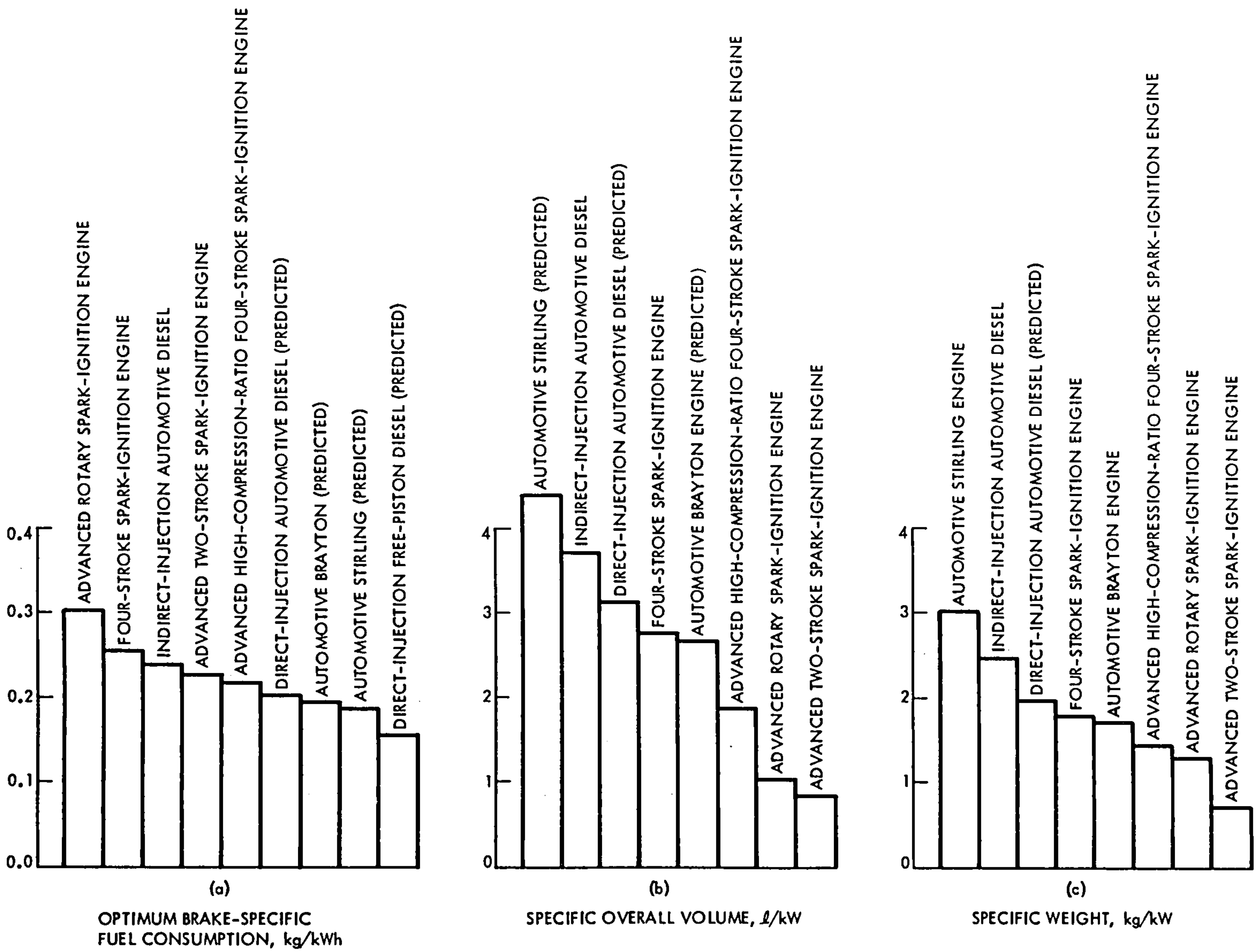


Figure 3-1. Comparison of Engine Design Criteria

engine will be available within the 10-year time frame under consideration for this study.

B. STIRLING ENGINE

Superior fuel efficiency, low NO_x emissions, low noise level, and the capabilty to adapt to a variety of alternative fuels are attractive features for any application in which an inherent relatively slow response and bulkiness do not matter. Because packaging considerations are of primary importance in hybrid-vehicle design, the Stirling engine does not seem to be suited for a hybrid powertrain concept where most (two-thirds) of the power must still be generated by the heat engine. The necessity of using hydrogen and relatively high operating pressures (>2000 psi) as well as the hazard of hydrogen leakage are of great concern in the use of a Stirling engine within the confines of a passenger car. In addition, it takes a minimum of 30 s for a cold Stirling engine to develop full power, which precludes its use in a rapid stop-start manner.

A more practical use of the Stirling concept is as a sealed free-piston engine linear-alternator unit to charge batteries (see Figure 2-26). This application would eliminate external leakage (the sealing problem with dynamic seals) and result in a better configuration for packaging. Approaches in this direction merit special attention although it is believed that they will have no bearing on any hybrid development work that will take place within the projected 10-year time frame.

C. DIESEL ENGINE

Compared with other candidate internal-combustion advanced engines (Figure 3-1), the indirect-injection diesel is less efficient in all categories, including that of fuel efficiency. If the problem of high-speed direct injection can be resolved, an improvement of about 15% over the indirect-injection engine would result. However, the diesel engine's inherently poor start-up ability in cold weather (requiring preheating) and its high starting torque and motoring power requirements would still deter its use in a hybrid vehicle. Idling would have to be permitted instead of a cutoff. Although the idle fuel-flow rate of diesels is only 20 to 25% that of spark-ignition engines, this characteristic reduces their actual fuel efficiency to some degree in hybrid operations. The idling diesel is an attractive option when viewed from the practical standpoint. However, hybrid accessory power and heating are still unresolved problems with heat-engine, stop-start operations.

D. HIGH-COMPRESSION, FOUR-STROKE, SPARK-IGNITION PISTON ENGINE

High-compression, four-stroke, spark-ignition piston engines are most likely to penetrate the market in the second half of the 1980s. The advanced four-stroke engine will satisfy hybrid-vehicle requirements in fuel economy, weight, and size (see Figure 3-1). A number of disadvantages and inherent problems with high-compression-ratio engines, however, should be taken into

consideration when weighing the four-stroke concept against other engine concepts.

Comparing an advanced engine having a compression ratio of 15:1 with the conventional engine currently in use in hybrid test vehicles (8.5:1), the torque required to crank the engine increases approximately by a factor of 1.5 or more (Figure 3-2). The service life of high-compression engines is also more affected by lubrication deficiencies, which are present in hybrid engines that must operate under frequent stop-start conditions. At this time it is still uncertain how frequent stop-start operations affect the wear pattern of conventionally designed, four-stroke gasoline engines in the long run. Although no evidence has been produced to date, Volkswagen (see Reference 1-1) has been keeping a low profile with their stop-start R&D work because of the overwhelming complex of open questions that must still be answered to satisfy product reliability and service-life guarantee requirements. The concensus among conservative engine designers is that the lubrication system must be specifically designed and developed to cope reliably with frequent and prolonged stop-start operations under worst-case environmental and driving conditions. For example, an electrically driven oil pump that allows the oil system to be pressurized prior to engine turnover or keeps the oil system pressurized all the time is believed necessary.

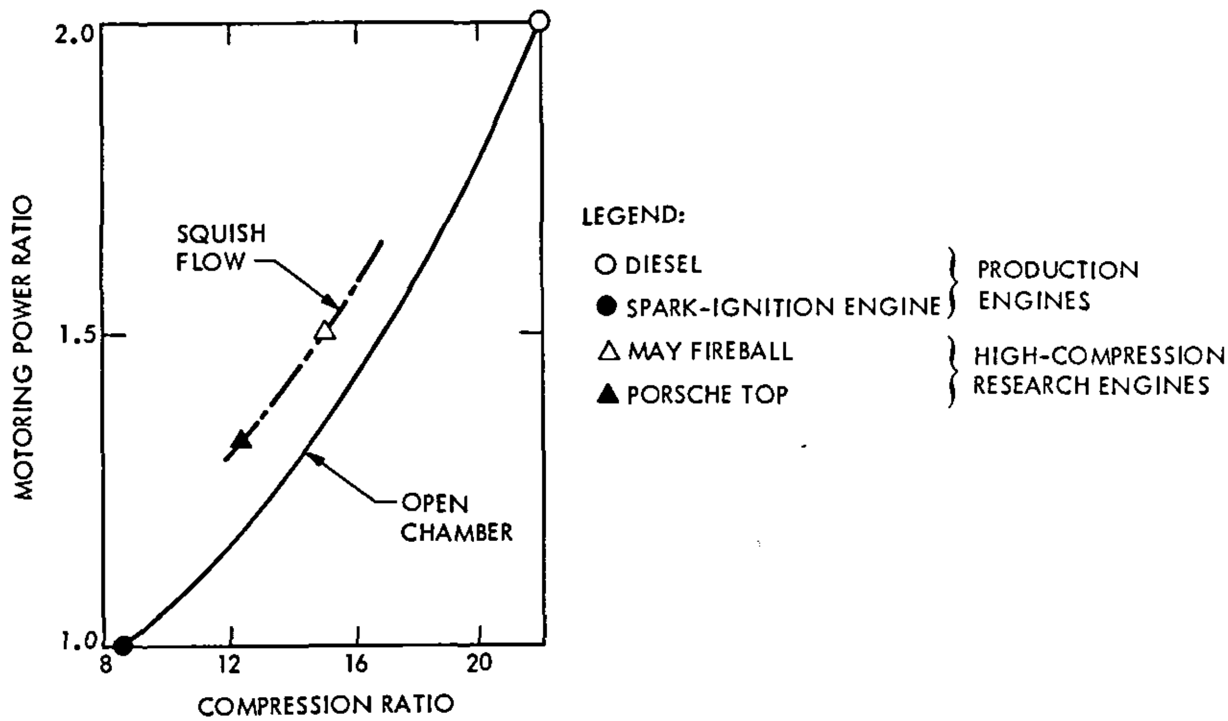


Figure 3-2. Estimate of Motoring Power Requirements Relative to Conventional 8.5:1 Compression-Ratio, Four-Stroke Engine

The NO_x emission must also be taken into account. High-compression, four-stroke engines are higher in NO_x than low-compression engines and depend more on efficient catalytic converters to meet emission requirements. Unfortunately, as discussed earlier and shown in Figure 1-2, the converter rarely achieves its best possible efficiency under frequent stop-start operating conditions.

Another factor to consider is that all recent four-stroke engine developments, including rotaries, use a continuous-flow, fuel-injection system like the Bosch K-Jetronic (or similar) system. As discussed earlier (see Reference 1-2), tests with hybrid vehicles using a low-compression, four-stroke engine equipped with a continuous fuel-flow injection system have exhibited undesirably high HC emission transients during engine start and CO-emission spikes during engine shutdown. The system's inherent tendency to overshoot at a sudden opening of the throttle produces a fuel-rich mixture. At engine cutoff, the manifold vacuum is broken to affect fuel cutoff by opening of a special (VW-introduced) vent valve. But with the engine still turning over for a few times before it comes to a full stop, the system remains partially open, and small amounts of fuel dribble into the engine until the system pressure has decayed below nozzle-opening pressure. This causes a lean mixture and high CO emission during every engine cutoff. The built-in, fuel-enrichment features that provide for a timed injection of extra fuel to facilitate cold start are also responsible for the high HC emission during engine start.

These deficiencies can be eliminated but will require special development. Instead of developing a stop-start, transient, emission-proof, continuous-flow, fuel-injection system, it would be more straightforward to use a pulse-flow system that controls fuel flow directly at the injection nozzle with a solenoid valve. An example is the Bosch K-Jetronic system. Based on the analytical studies conducted at General Electric (see Reference 1) a K-Jetronic-equipped, 1.7-l, four-stroke, spark-ignition production Audi engine was chosen for use in experimental hybrid vehicles because it was available and because the developer (Volkswagen) already had some stop-start test experience with this engine. The aforementioned vent valve that breaks the manifold vacuum at engine cutoff was added later to stop the fuel flow faster at engine cutoff.

E. TWO-STROKE, SPARK-IGNITION PISTON ENGINE

For the near future, the advanced cylinder, fuel-injected, two-stroke, spark-injection piston engine seems to be the most attractive engine concept for use in small automobiles and hybrid vehicles. It has fewer moving parts, is mechanically simple, is most tolerant to poor fuel quality, and ranks highest among all other concepts in service needs and reliability. Its fuel efficiency (see Figures 2-7, 2-8, and 3-1a) is competitive with high-compression, four-stroke, spark-ignition engines and is considerably better than that of current four-stroke engines, including the indirect-injection diesel engine.

The relatively large amount of residual gases still present at the start of each new cycle represents a built-in EGR that keeps NO_x emissions low (see

Figure 3-1a), despite the fact that combustion temperatures are higher than those encountered in four-stroke engines. The higher combustion and exhaust temperatures of the two-stroke engines help to burn off HC emissions to an extent that makes the use of a separate external thermal reactor almost unnecessary (see Figures 2-3 and 3-1a). The volume of a tuned two-stroke exhaust system is large enough to accomplish an efficient burn-off of unburned hydrocarbons. The two-stroke, low speed, and idling instability that has been considered inherent in the two-stroke engine has been largely eliminated (see Figure 2-11). The fuel-to-oil ratio of advanced two-stroke engines (see (Figure 2-12) ranges from 100 to 200, and the HC emissions generated by oil mixed into the airflow for lubrication (see Figure 2-10) is now extremely low.

The motoring requirements for two-stroke engines (Figure 3-3) are considerably lower than those for conventional four-stroke engines because there are no moving valves, and the compression ratio is low (7.5:1). This advantage can translate into subtantial energy savings in drive cycles that require frequent stop-start of the engine. Current hybrid drive-cycle analysis does not take differences in cranking power requirements into consideration when comparing engines. Another advantage of the two-stroke engine is that it requires a minimal warm-up period, and full power can be drawn from the engine upon demand. The striking weight and size advantages of the advanced two-stroke engine over other candidate engines are also significant (see Figure 2-6 and 3-1).

All of the engines discussed are liquid cooled. Liquid-cooled engine systems are usually slightly heavier but, as a system, they are more space-efficient and more versatile from the packaging standpoint than forced-air-

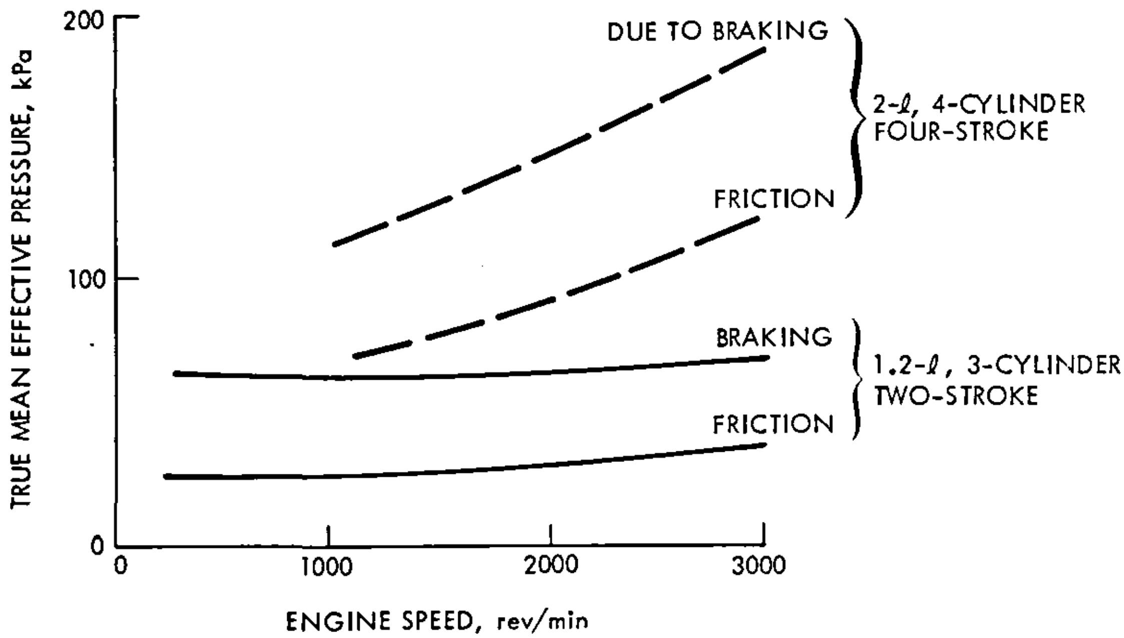


Figure 3-3. Comparison of Motoring Mean-Effective Pressures of Two- and Four-Stroke Spark-Ignition Engines (see Reference 2-9)

cooled engine systems. Liquid-cooled engines are also more ($\approx$10%) fuel-efficient than air-cooled engines. Forced-air cooling, however, seems to be the better choice from the reliability and cost standpoint, particularly in case of an auxiliary heat engine for a primary electric car.

F. FOUR-STROKE, SPARK-IGNITION ROTARY ENGINE

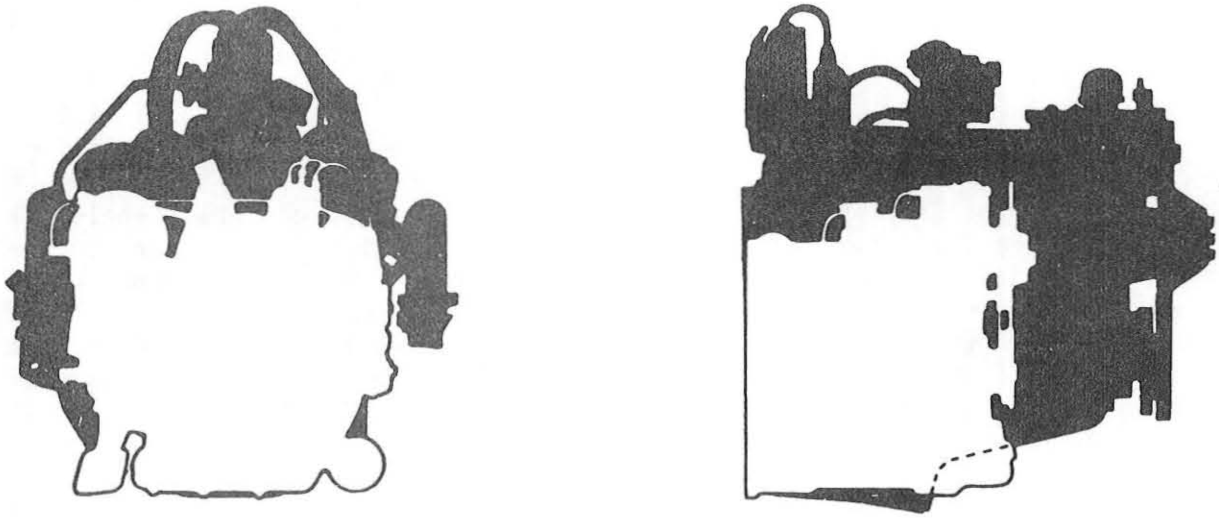
Despite the fact that the fuel economy of the advanced rotary engine is not competitive with advanced two- and four-stroke engines, the rotary engine concept is still an attractive option for hybrid application because of its speed, weight, and size advantages (see Table 2-1, and Figures 3-1 and 3-4). However, if a single-rotor configuration must be chosen to meet hybrid-vehicle power requirements, the packaging and weight advantages of the rotary engine over its competitors are greatly reduced (Figure 3-5) because of larger bearings and balancing weights.

As shown in Figure 3-6, the motoring requirements for rotary engines are considerably lower (up to 50%) than those of four-stroke piston engines. Because it is effectively mixture-lubricated in critical wear areas, the fuel-injected rotary engine requires little, if any, warmup before full power can be demanded. Another positive feature is its inherent potential for smoothness and speed (see Figure 2-1), which offers the possibility of an integral single-shaft design with a high-speed alternator. A shorter service life than piston engines may be acceptable for hybrid application because the engine works only part of the total time the vehicle is driven. Advanced rotary engines are expected to be more costly than other advanced candidate engines. However, the cost factor may be acceptable in a hybrid vehicle if the advantages it has to offer are of overriding importance.

G. FREE-PISTON ENGINE

The integration of a free-piston engine with a linear alternator is potentially attractive for battery charging in hybrid vehicles because of its compactness and smooth operation. Because of its capability of producing high compression ratios with relatively low friction losses, the free-piston concept lends itself favorably to the implementation of diesel as well as spark-ignited (Otto-like) combustion with alternative fuels that tolerate high compression ratios, such as alcohols and ammonia. Squish techniques as applied to high-compression gasoline crankshaft engines cannot be applied because the piston stroke and the points of piston reversal are not fixed in a free-piston engine. The free-piston diesel has a superior start capability because the piston stroke is not limited, and a high compression ratio can be produced by electrically exciting the oscillating masses into resonance. The idle flow rate should also be considerably lower than that of crankshaft engines because less friction must be overcome for minimum sustained operation at zero load.

Many of the inherent problems of the free-piston, internal-combustion engine of conventional design that limit its potential to compete with crankshaft engines can be overcome with modern technology. For example, the complex fuel-injection and timing requirements needed for optimum free-piston



LEGEND:

BLACK - V8 PISTON ENGINE

WHITE - ROTARY ENGINE

Figure 3-4. Size Comparison of Twin Rotor Wankel with V-8 Piston Engine of Equivalent Performance (Reference 3-1)

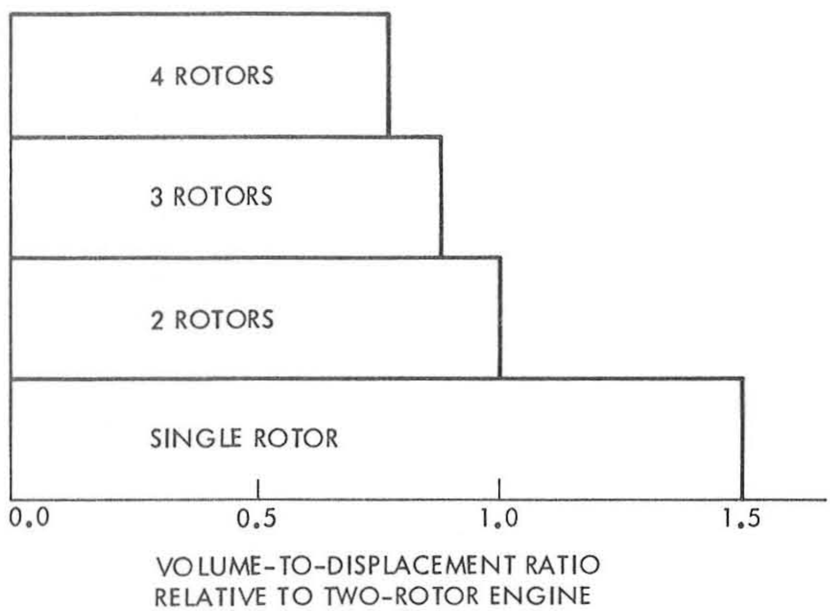


Figure 3-5. Effect of Number of Rotors on Overall Volume to Displacement Ratio of Rotary Engine Relative to Two-Rotor Design (Reference 3-2).

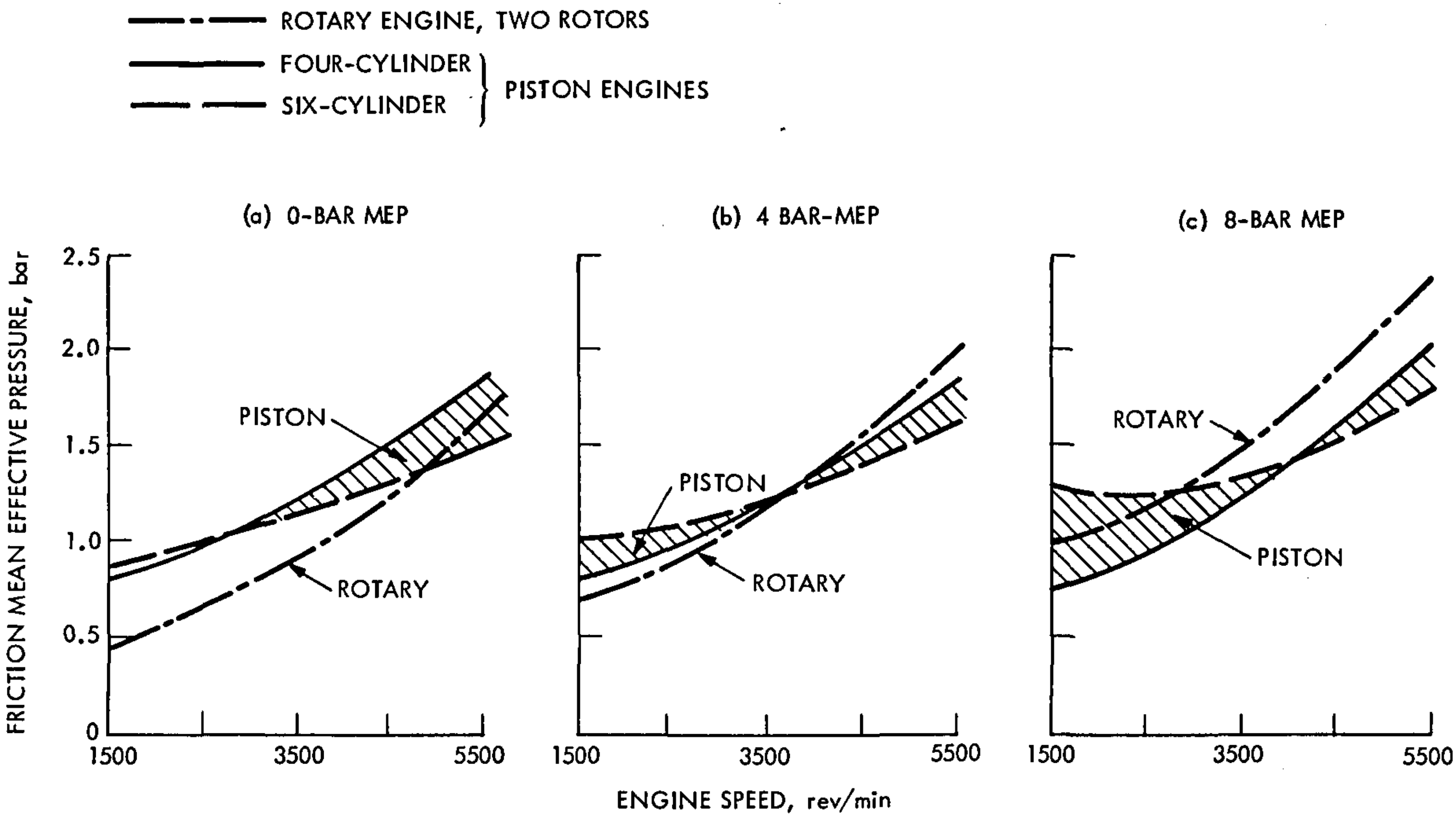
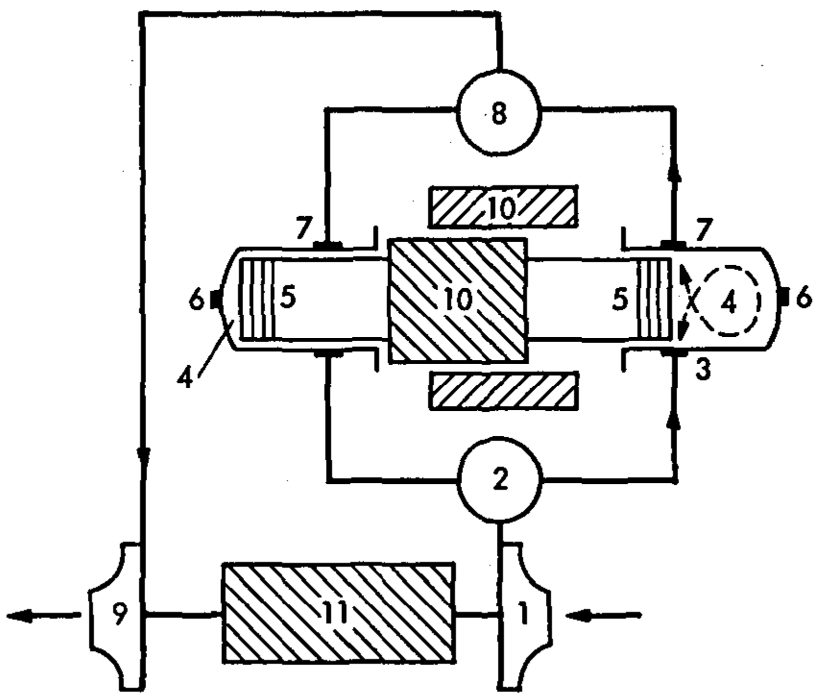


Figure 3-6. Comparison of Rotary Engine Friction Mean-Effective Pressure with Four-Stroke Piston Engine Data (Reference 3-3)

operation can be met today with electronic techniques to sense piston speed and reversal points, and to ensure resonance at various load levels by controlling fuel-injection timing, fuel flow, and pressure in the bounce chamber. The application of trip- and miss-control techniques is also possible.

Piston heating, which has been a problem with small, high-speed, free-piston engines, can be overcome with modern high-temperature metallic materials and, perhaps, with ceramics. The free-piston engine lends itself ideally to the application of ceramic materials and gas lubrication because of its simple configuration, its relatively low dynamic forces, and the absence of lateral force action on the piston. For battery charging in hybrid passenger cars (15-20 kW), a single-mass, turbocompounded, ceramic, free-piston diesel or alcohol-fueled Otto engine with opposing cylinders (Figure 3-7) represents an attractive concept from the fuel efficiency, weight, and packaging standpoint.

As previously mentioned, the free-piston, linear-alternator concept is also an attractive option for the Stirling engine (see Figure 2-26). It would eliminate the external sealing problem with high-pressure hydrogen through dynamic seals and make the Stirling concept more acceptable for hybrid battery charging as far as packaging and safety are concerned.



LEGEND:

1 - BLOWER
2 - INLET PLENUM
3 - INLET PORTS
4 - COMBUSTION SPACE
5 - PISTONS
6 - FUEL INJECTION NOZZLE
SPARK PLUG (ALCOHOL ENGINE)
7 - EXHAUST PORTS
8 - EXHAUST PLENUM
9 - TURBINE
10 - LINEAR ALTERNATOR, STARTER SOLENOID
11 - ALTERNATOR, BLOWER STARTER MOTOR

Figure 3-7. Schematic of Turbocompounded Free-Piston Power Unit Concept

SECTION IV

SUMMARY AND CONCLUSION

Based on the information presented and considering the self-imposed rating criteria and restraints for the selection of hybrid-vehicle engines, the study concludes as follows:

(1) Among the advanced engine concepts under consideration, the water-cooled cylinder fuel-injected, crankcase-scavenged, two-stroke, spark-ignition piston engine emerges as the best choice for future hybrid vehicles of the parallel powertrain concept. Because of its low-compression ratio and large internal turbulence this engine is tolerant of poor quality (low-octane) fuels. The two-stroke inherent lubrication physics prevent long-term wear damage due to stop-start operations. Its specific fuel consumption and the power required to crank the engine are lowest. The number of firings encountered per unit cranking energy and the chance to fire up during a given number of turns are highest. In summary, the fuel-injected, two-stroke engine is most cost- and energy-efficient, and it satisfies all the other HV needs. A carbureted, air-cooled version of an advanced two-stroke engine equipped with an up-to-date oil-spray lubrication system may satisfy the requirements for an auxiliary engine for a primarily electrically driven vehicle, depending upon area emission restraints.

(2) The spark-ignition, four-stroke engine of current low-compression design is the best backup choice for a hybrid vehicle of the parallel powertrain concept. However, it should be equipped with a fuel-injection system that is positively controlled at the point of injection to reduce to a minimum transient emissions due to dribbling after engine cutoff. It should also be equipped with an electrical, externally powered oil pump that allows pressurization of the lubrication system prior to engine start. This should eliminate the concern about damages due to long-term wear effects that potentially result from frequent stop-start operation.

(3) Rotary four-stroke, spark-ignition engines are not competitive in regard to fuel economy, but their small-volume and high-speed capability may be a decisive factor where the need for these properties exists. The rotary engine's high-speed capability permitting it to run directly coupled with a high-speed alternator, as well as its low starting and motoring power requirements, should translate into fuel savings that must be taken into account when the rotary engine is compared with four-stroke engines in an analysis of hybrid vehicle performance.

(4) The high-compression, four-stroke, spark-ignition engine is not well-suited for hybrid operations requiring stop-start capability because of the inherent problems associated with high-compression-ratio engines. These are mainly high starting torque and energy

losses due to high cranking power requirements, high NOx emissions in combination with a low catalytic converter efficiency, and an increased susceptibility to the lubrication deficiencies brought about by stop-start operations.

(5) The indirect-injection automotive diesel is not competitive at all for hybrid vehicle use. The direct-injection diesel could be an alternative to the spark-ignition, two-stroke engine in applications in which its size and weight do not matter and when it is allowed to idle instead of requiring an engine cutoff. An idling DI-diesel may have advantages over the two-stroke engine where prolonged air conditioning and heating times are involved.

(6) A free-piston, direct-injection diesel or a spark-ignition engine fueled with alcohol and coupled directly to a linear alternator are attractive for future use in battery charging, from the standpoint of packaging, fuel efficiency, low vibration, and noise.

(7) Because of the inherent physical laws governing external combustion engines such as the Brayton, Stirling, and steam engines, they are not suited for use in hybrid-vehicle, stop-start operations at all without special measures that remain to be developed. The large number of still-unresolved basic development problems as well as an uncertain funding situation in this area of research precludes the application of external combustion engines for hybrid uses within the considered (10-year) time period.

SECTION V

RECOMMENDATIONS

This study recommends follow-on analysis and experimentation with candidate engine concepts and components that are necessary to support a final selection of a heat-engine concept for future hybrid vehicles. Heat engines are usually not conceived and developed to meet the specific requirements of hybrid-vehicle operation. Heat engines must be adapted to meet these requirements. Developers usually cannot provide all the data necessary to evaluate a heat-engine concept in regard to its potential for hybrid use and to assess the modifications necessary. Specific hybrid-related analysis and tests other than vehicular tests alone are therefore needed to generate the data base necessary for hybrid-engine selection and development planning. Listed in order of priority are the following recommendations resulting from this study.

(1) Conduct a comparative driving cycle analysis with most promising advanced engine concepts, including an assessment of the transient energy losses resulting from the motoring, inertial, and starting characteristics of each individual concept known to date. Current hybrid-vehicle models do not realistically account for transient energy losses because of high-speed motoring, delayed starts, and clutching during frequent and rapid stop-start operations.

(2) Conduct evaluation bench tests with the high-speed, fuel-injection systems discussed in this report to verify the claims made by their developers, to understand how they function and to determine how to assess their true potential for hybrid use.

(3) Simulate hybrid typical heat-engine operations on the dynamometer. The capability of determining transient power output, cranking power input, fuel flow, and emissions must be provided. Listed in order of priority, the following engine concepts should be tested:

   (a) A low-compression, four-stroke, spark-ignition engine (Audi 1.7) identical to those installed in the hybrid vehicles currently tested. This is necessary to better understand the transient emission behavior of the vehicles tested and to take measures to reduce the transient emission of the vehicles using this engine to a possible minimum. A test program of this nature that uses hardware and equipment on hand was proposed in September 1982 (Reference 5-1).

   (b) A two- or three-cylinder, two-stroke Marine engine developed in the United States modified to meet the combustion and fuel-injection requirements of the OEC research engine (see Reference 2-10) discussed in subsection II-A3. The engine should be equipped initially with a conventional dual plunger Bosch-jerk pump, unless suggested otherwise as the result of the bench tests described in V-(2). Two of the sources

contacted (see References 2-7 and 2-11) have offered to conduct a task of this nature at cost with their own research hardware.

(c) A fuel-injected rotary engine has been recently marketed by Mazda that develops 100 kW (130 hp). The power of this engine is more than twice the power output projected for hybrid engines, but data could be reduced to single-rotor configuration after the basic friction and pumping losses of the two-rotor engine have been determined.

(d) A high-compression, spark-ignition, four-stroke engine. A 2-l engine with a 13:1 compression ratio may possibly be made available from the discussed sources after their own tests with this engine are completed.

(4) Conduct analysis as described in V-(1). Use the transient data generated in tests recommended in V-(3).

(5) Conduct a project design study of an alcohol-fueled spark-ignited, free-piston engine coupled to a linear alternator. Use of the latest technology and innovative design concepts.

(6) Keep abreast with ongoing developments through literature and personal communications. Considerable effort has been made in the course of this study to locate and approach competent sources and to encourage their cooperation. It is important to maintain these contacts. The evaluation and demonstration of foreign advanced heat-engine technology represents a unique opportunity to gain experience and to make contributions to the solution of pacing problems for the benefit of the DOE Hybrid Vehicle Program as well as for United States industry in general.

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APPENDIX A

CONTACTS MADE IN CONJUNCTION WITH THE HYBRID VEHICLE HEAT-ENGINE EVALUATION TASK

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